American Society *of* Heating *and* Ventilating Engineers

Guide, 1934

An Instrument of Service prepared for the Profession—and Containing reference data on the design and specification of heating and ventilating systems—Based on the Transactions—the Investigations of the Research Laboratory and Cooperating Institutions—and the Practice of the Members and Friends of the Society

together with a

Manufacturers' Catalog Data Section Containing Essential and Reliable Information Concerning Modern Equipment

also

The Roll of Membership of the Society

with

Complete Indexes to Technical and Catalog Data

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Dedicated
To the Advancement of
The Profession
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Its Allied Industries

Contents

PREFACE TO THE 12th EDITION

THE Technical Data Section of this, the 12th Edition of the A.S.H.V.E. Guide, has been enlarged to include newly developed data that are vitally important in meeting the present day demands of engineers who devote their time to heating, ventilating and air conditioning practice. From the practical experience of members as well as from available research sources useful facts have been gathered and incorporated in the 42 chapters which have been arranged for convenient reference.

Since its first appearance in 1922 The Guide has maintained its leadership in the development of the heating and ventilating art so that today it is the recognized authority of the profession and industry. It influences thousands of engineers, architects, contractors and students who are designing, operating, specifying, installing and studying systems and apparatus, the functions of which are to create comfort and to improve the efficiency of processing.

All of the data in the previous edition have been reviewed, many chapters have been revised and amplified while others have been completely replaced. The new chapters include the Cooling Load and Cooling Methods, Unit Conditioners, Radiant and Electric Heating, Humidifying and Dehumidifying Equipment, Steam Heating Systems and Piping. Extensive changes will be noted in the chapters on Industrial Air Conditioning, Natural Ventilation, Central Fan Systems, Air Cleaning Equipment, Sound Control, Mechanical Furnace Systems, Radiators and Gravity Convectors, Heating Boilers, Pipe Insulation, Pipe, Fittings and Welding, Definitions and Terms. The remaining chapters were revised in order to bring them up-to-date.

Slight modifications were made in the chapters on Heat Transmission and Air Filtration. The transmission and air leakage values introduced in the 1933 edition have been retained although considerable comment came from users because some of the factors were greater than those used in The Guide 1932. However, a careful check of the experimental work which produced the basic figures for The Guide 1933 indicated that the values determined by the research groups responsible for these factors were reliable and that the values should be used. However, the method of application in practice has been slightly modified in this 1934 edition of The Guide.

With 42 chapters and a comprehensive index totaling 620 pages, The Guide 1934 presents the largest technical data section ever compiled and published by the American Society of Heating and Ventilating Engineers. Supplemented by the engineering data of leading manufacturers in the Catalog Section, the user will find essential facts that are invaluable in laying out a plant or selecting equipment. This unique method effectively covering both practice and equipment makes The Guide outstanding in its service to the user.

Leading manufacturers of equipment recognize this fact and find that catalog advertising is most effective in extending the use of Modern Equipment in this highly specialized and progressive field. By means of this cooperation, the American Society of Heating and Ventilating Engineers can produce and distribute The Guide economically and thereby contribute effectively toward the general advancement of the profession and its allied industries.

The Guide 1934 is released for service and it is the desire of The Guide Publication Committee that it will perform its intended mission for its thousands of friends in the profession.

W. L. Fleisher, *Chairman*
GUIDE PUBLICATION COMMITTEE

EDITORIAL ACKNOWLEDGMENT

FOR the important work of creating The Guide 1934 the assistance of engineers of specialized training and knowledge was enlisted. The careful reviewing, compiling and editing of data selected from authoritative sources was accomplished through the cooperative efforts of the following men who have unselfishly given of their time and knowledge:

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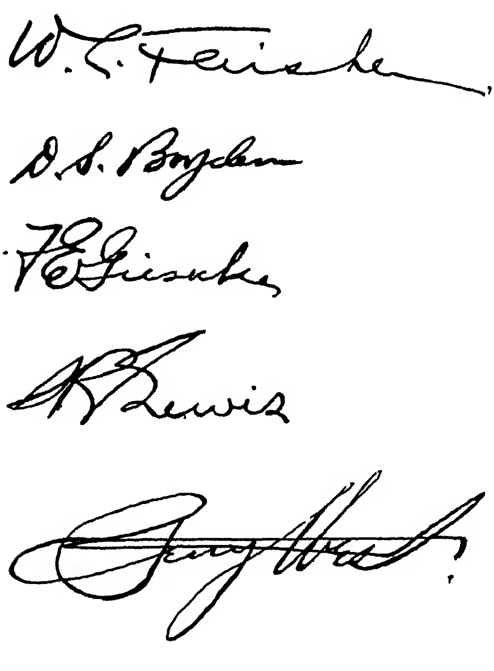

, *Chairman*

CODE *of* ETHICS *for* ENGINEERS

ENGINEERING work has become an increasingly important factor in the progress of civilization and in the welfare of the community. The engineering profession is held responsible for the planning, construction and operation of such work and is entitled to the position and authority which will enable it to discharge this responsibility and to render effective service to humanity.

That the dignity of their chosen profession may be maintained, it is the duty of all engineers to conduct themselves according to the principles of the following Code of Ethics:

1—The engineer will carry on his professional work in a spirit of fairness to employees and contractors, fidelity to clients and employers, loyalty to his country and devotion to high ideals of courtesy and personal honor.

2—He will refrain from associating himself with or allowing the use of his name by an enterprise of questionable character.

3—He will advertise only in a dignified manner, being careful to avoid misleading statements.

4—He will regard as confidential any information obtained by him as to the business affairs and technical methods or processes of a client or employer.

5—He will inform a client or employer of any business connections, interests or affiliations which might influence his judgment or impair the disinterested quality of his services.

6—He will refrain from using any improper or questionable methods of soliciting professional work and will decline to pay or to accept commissions for securing such work.

7—He will accept compensation, financial or otherwise, for a particular service, from one source only, except with the full knowledge and consent of all interested parties.

8—He will not use unfair means to win professional advancement or to injure the chances of another engineer to secure and hold employment.

9—He will cooperate in upbuilding the engineering profession by exchanging general information and experience with his fellow engineers and students of engineering and also by contributing to work of engineering societies, schools of applied science and the technical press.

10—He will interest himself in the public welfare in behalf of which he will be ready to apply his special knowledge, skill and training for the use and benefit of mankind.

Chapter 1

THERMODYNAMICS OF AIR CONDITIONING

Dalton's Law, Dry- and Wet-Bulb Temperatures, Partial Pressures, Dew-Point Temperature, Humidity, Relation of Dew-Point to Relative Humidity, Adiabatic Saturation of Air, Total Heat and Heat Content, Energy Equation for Air Conditioning Processes, Psychrometric Chart, Examples in Use of Chart, Rate of Evaporation

THE subject of air conditioning involves a study of the desirable or necessary atmospheric conditions for human comfort or for manufacturing processes, as the case may be, and of the various physical or thermodynamic relationships of water vapor and the gases with which it is mixed. This chapter deals with the latter.

DALTON'S LAW

Air is a mixture of a number of dry gases and water vapor. The percentage of the gases contained in air remains relatively constant, and is usually given no consideration by the air-conditioning engineer. The weight of vapor mixed with dry air varies over wide limits, and this percentage affects the health and feeling of warmth of man, and the behavior of many materials in the process of manufacture.

A mixture of dry gases and water vapor, such as atmospheric air, obeys Dalton's Law of Partial Pressures; each gas or vapor in a mixture, at a given temperature, contributes to the observed pressure the same amount that it would have exerted by itself at the same temperature had no other gas or vapor been present. If p = the observed pressure of the mixture and p_1, p_2, p_3, etc. = the pressure of the gases or vapors corresponding to the observed temperature, then

$$p = p_1 + p_2 + p_3, \text{ etc.}$$

DRY- AND WET-BULB TEMPERATURES, PARTIAL PRESSURES

Air is said to be saturated at a given temperature when the water vapor mixed with the air is in the dry saturated condition, or what is the equivalent, when the space occupied by the mixture holds the maximum possible weight of water *vapor* at a given temperature. If the water vapor mixed with the dry air is superheated, *i.e.*, if its temperature is above the temperature of saturation for the actual water vapor partial pressure, the air is not saturated.

The starting point of most applications of thermodynamic principles to air-conditioning problems is the experimental determination of the dry-bulb and wet-bulb temperatures, and sometimes, the barometric pressure.

The *dry-bulb temperature* of the air is the temperature indicated by any type of thermometer not affected by the water vapor content or relative humidity of the air. The *wet-bulb temperature* is determined by a thermometer with its bulb encased in a fine mesh fabric bag moistened with clean water and whirled through the air until the thermometer assumes a steady temperature. This steady temperature is the result of a dynamic equilibrium between the rate at which heat is transferred from the air to the water on the bulb and the rate at which this heat is utilized in evaporating moisture from the bulb. The rate at which heat is transferred from the air to the water is substantially proportional to the wet-bulb depression $(t - t')$, while the rate of heat utilization in evaporation is proportional to the difference between the saturation pressure of the water at the wet-bulb temperature and the actual partial pressure of the water vapor in the air $(e' - e)$. Carrier's equation for this dynamic equilibrium is

$$\frac{e' - e}{t - t'} = \frac{B - e'}{2800 - 1.3t'} \tag{2a}$$

In the form commonly used,

$$e = e' - \frac{(B - e')\,(t - t')}{2800 - 1.3t'} \tag{2b}$$

where

e = actual partial pressure of water vapor in the air, in inches of mercury.
e' = saturation pressure at wet-bulb temperature, in inches of mercury.
B = barometric pressure, in inches of mercury.
t = dry-bulb temperature, in degrees Fahrenheit.
t' = wet-bulb temperature, in degrees Fahrenheit.

Formula 2b may be used to determine the actual partial pressure of the water vapor in a dry air-water vapor mixture. Then, from Dalton's Law of Partial Pressures, Equation 1, it follows that the partial pressure of the dry air is $(B - e)$.

If a mixture of dry air and water vapor, initially unsaturated, be cooled at constant pressure, the temperature at which condensation of the water vapor begins is called the *dew-point temperature*. Clearly the dew-point is the saturation temperature corresponding to the actual partial pressure, e, of the water vapor in the mixture.

HUMIDITY

Humidity is the water vapor mixed with dry air in the atmosphere. *Absolute humidity* has a multiplicity of meanings, but usually the term refers to the weight of water vapor per unit volume of space occupied, expressed in grains or pounds per cubic foot. With this meaning, absolute humidity is nothing but the actual density of the water vapor in the mixture and might better be so called. A study of Keenan's Steam Tables[1] indicates that water vapor, either saturated or super-heated, at partial pressures lower than 4 in. of mercury may be treated as a gas with

[1]Published by *American Society of Mechanical Engineers*, see abstract Chapter 41.

a gas constant R of 1.21 in the characteristic equation of the gas $pV = wR\ (t + 460)$. Within such limits, the density (δ) of water vapor is

$$\delta = \frac{w}{V} = \frac{e}{1.21\ (t + 460)} \text{ (pounds per cubic foot)} \tag{3a}$$

$$= \frac{5785\ e}{t + 460} \text{ (grains per cubic foot)} \tag{3b}$$

where

e = actual partial pressure of vapor, in inches of mercury.
t = dry-bulb temperature, in degrees Fahrenheit.

Another meaning sometimes given to absolute humidity is the weight of water vapor mixed with a unit weight of dry air. This quantity is the ratio of the density of the vapor to the density of the dry air, and since the gas constant R for dry air is 0.753, the weight of water vapor mixed with 1 lb of *dry air* is

$$W = \frac{e}{1.21\ (t + 460)} \div \frac{B\text{-}e}{0.753\ (t + 460)}$$

$$= 0.622 \left(\frac{e}{B\text{-}e}\right) \text{ (pounds)} \tag{4a}$$

$$= 4354 \left(\frac{e}{B\text{-}e}\right) \text{ (grains)} \tag{4b}$$

where

e = actual partial pressure of vapor, in inches of mercury.
B = total pressure of mixture (barometric pressure), in inches of mercury.

Relative Humidity

Relative humidity (Φ) is either the ratio of the actual partial pressure (e) of the water vapor in the air to the saturation pressure (e_t) at the dry-bulb temperature, or the ratio of the actual density (δ) of the vapor to the density of saturated vapor (δ_t) at the dry-bulb temperature. That is:

$$\Phi = \frac{e}{e_t} = \frac{\delta}{\delta_t} \tag{5}$$

Relative humidity, so defined, is not exactly equal to the ratio of the weight of vapor per *pound* of dry air (W) to the weight of saturated vapor per pound of dry air (W_t). This quantity is sometimes called per cent humidity, for from Equations 4 and 5,

$$\frac{W}{W_t} = 0.622 \left(\frac{\Phi\ e_t}{P - \Phi\ e_t}\right) \div 0.622 \left(\frac{e_t}{B\text{-}e_t}\right) = \frac{\Phi\ (B\text{-}e_t)}{B - \Phi e_t} \tag{6}$$

It is not *exactly* correct, therefore, to find the weight of vapor mixed with each pound of dry air (W) by multiplying the weight of vapor mixed with each pound of dry air for saturation at the dry-bulb temperature (W_t) by the relative humidity (Φ), although the *error usually is small*, particularly if the relative humidity is high.

With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and dew-point temperatures are equal. With a relative humidity less than

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With a relative humidity of 100 per cent, the dry-bulb, wet-bulb, and dew-point temperatures are equal. With a relative humidity less than

100 per cent, the dry-bulb exceeds the wet-bulb, and the wet-bulb exceeds the dew-point temperature.

RELATION OF DEW-POINT TO RELATIVE HUMIDITY

A peculiar relationship exists between the dew-point and the relative humidity and this is found most useful in air conditioning work. This is, that for a fixed relative humidity there is substantially a constant difference between the dew-point and the dry-bulb temperature over a considerable temperature range. Table 1 giving the dry-bulb and dew-point temperatures, and dew-point differentials for 50 per cent relative humidity, illustrates this relationship clearly.

TABLE 1. DRY-BULB AND DEW-POINT TEMPERATURES FOR 50 PER CENT RELATIVE HUMIDITY

Dry-bulb temperature	65.0	70.0	75.0	80.0	85.0	90.0
Dew-point temperature	45.8	50.5	55.25	59.75	64.25	68.75
Difference between dew-point and dry-bulb temperature	19.2	19.5	19.75	20.25	20.75	21.25

It will be seen from an inspection of this table that the difference between the dew-point temperature and the room temperature is approximately 20 deg throughout this range of dry-bulb temperatures or, to be more exact, the differential increases only 10 per cent for a range of practically 25 deg.

This principle holds true for other humidities and is due to the fact that the pressure of the water vapor practically doubles for every 20 deg through this range.

The approximate relative humidity for any difference between dew-point and dry-bulb temperature may be expressed in per cent as:

$$\frac{100}{2^{\frac{t-t_1}{20}}} \qquad (7)$$

where

t_1 = dew-point temperature.

This principle is very useful in determining the available cooling effect obtainable with saturated air when a desired relative humidity is to be maintained in a room, even though there may be a wide variation in room temperature. This problem is one which applies to certain industrial conditions, such as those in cotton mills, tobacco factories, etc., where relatively high humidities are carried and where one of the principal problems is to remove the heat generated by the machinery. It also permits the use of a differential thermostat, responsive to both the room temperature and to the dew-point temperature, to control the relative humidity in the room.

Table 2 gives, for different temperatures, the density of saturated vapor (δ_t), the weight of saturated vapor mixed with 1 lb of dry air (W_t), (for a relative humidity of 100 per cent and a barometric pressure (B) of

29.92 in. of mercury), the specific volume of dry air, and the volume of an air-vapor mixture containing 1 lb of dry air (for a relative humidity of 100 per cent and a pressure of 29.92 in. of mercury). The preceding equations or the data from Table 2 may be conveniently used in solving the following typical problems:

Example 1. Humidifying and Heating. Air is to be maintained at 70 F with a relative humidity of 40 per cent ($\Phi = 0.4$) when the outside air is at 0 F and 70 per cent relative humidity ($\Phi = 0.7$) and a barometric pressure (B) of 29.92 in. of mercury. Find the weight of water vapor added to each pound of dry air and the dew-point temperature of the humidified air.

Solution. From Equation 4a and Table 2,

$$W_1 = 0.622\left(\frac{0.7 \times 0.0375}{29.92 - 0.0263}\right) = 0.000547 \text{ lb per pound of dry air.}$$

$$W_2 = 0.622\left(\frac{0.4 \times 0.7386}{29.92 - 0.295}\right) = 0.00618 \text{ lb per pound of dry air.}$$

The water vapor added per pound of dry air must be $(W_2 - W_1)$ or 0.005633 lb. By inspection of Table 2, $W_t = 0.00618$ at 44.5 F, so this is the dew-point temperature of the humidified air.

An approximation of the same result from Table 2 is

$W_1 = 0.7 \times 0.000781 = 0.000547$ lb per pound of dry air.
$W_2 = 0.4 \times 0.01578 = 0.006312$ lb per pound of dry air.

The water vapor added per pound of dry air is approximately 0.005765 lb and the dew-point temperature is approximately 45 F. The degree of approximation is evident.

Example 2. Dehumidifying and Cooling. Air with a dry-bulb temperature of 84 F, a wet-bulb of 70 F, or a relative humidity of 50 per cent ($\Phi = 0.5$) and a barometric pressure (B) of 29.92 in. of mercury is to be cooled to 54 F. Find the dew-point temperature of the entering air and the weight of vapor condensed per pound of dry air.

Solution. From Equation 4a and Table 2,

$$W_1 = 0.622\left(\frac{0.5 \times 1.174}{29.92 - 0.587}\right) = 0.01245 \text{ lb per pound of dry air}$$

$$W_2 = 0.622\left(\frac{0.42}{29.92 - 0.42}\right) = 0.00887 \text{ lb per pound of dry air.}$$

Since $W_1 = W_t$ when $t = 63.3$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is $(W_1 - W_2)$ or 0.00358 lb per pound of dry air.

An approximate result is

$W_1 = 0.5 \times 0.02547 = 0.01274$ lb per pound of dry air.
$W_2 = 1 \times 0.00887 = 0.00887$ lb per pound of dry air, since the exit air is saturated.

Since $W_1 = W_t$ at $t = 64$ F, this is the dew-point temperature of the entering air. The weight of vapor condensed is 0.00387 lb per pound of dry air. The degree of approximation is again evident.

ADIABATIC SATURATION OF AIR

The process of adiabatic saturation of air is of considerable importance in air conditioning. Suppose that 1 lb of dry air, initially unsaturated but carrying W lb of water vapor with a dry-bulb temperature, t, and a wet-bulb temperature, t', be made to pass through a tunnel containing an exposed water surface. Further assume the tunnel to be completely insulated, thermally, so that the only heat transfer possible is that between

TABLE 2. MIXTURES OF AIR AND SATURATED WATER VAPOR[a]

Temp., F	[b]Pressure of Saturated Vapor: In. of Hg	[b]Pressure of Saturated Vapor: Lb per Sq In.	Weight of Saturated Vapor: per Cu Ft, Pounds	Weight of Saturated Vapor: per Cu Ft, Grains	Weight of Saturated Vapor: per lb of Dry Air, Pounds	Weight of Saturated Vapor: per lb of Dry Air, Grains	Volume in Cu Ft: of 1 lb of Dry Air	Volume in Cu Ft: of 1 lb of Dry Air + Vapor to Saturate it	Heat Content in Btu of 1 lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[c]Heat Content in Btu of 1 lb of Dry Air with Vapor to Saturate it
0	0.0375	0.0184	0.0000674	0.472	0.000781	5.47	11.58	11.59	0.0	0.852	0.852
2	.0417	.0204	.0000746	.522	.000869	6.08	11.63	11.65	0.482	0.946	1.428
4	.0462	.0227	.0000823	.576	.000963	6.74	11.68	11.70	0.964	1.047	2.011
6	.0512	.0252	.0000909	.636	.001067	7.47	11.73	11.75	1.446	1.159	2.605
8	.0567	.0279	.0001001	.701	.001183	8.28	11.78	11.80	1.928	1.285	3.213
10	0.0628	0.0308	0.0001103	0.772	0.001309	9.16	11.83	11.86	2.411	1.420	3.831
12	.0694	.0341	.000121	.850	.001447	10.13	11.88	11.91	2.893	1.568	4.461
14	.0766	.0376	.000134	.935	.001599	11.19	11.94	11.97	3.375	1.731	5.106
16	.0846	.0415	.000147	1.028	.001764	12.35	11.99	12.02	3.858	1.908	5.766
18	.0932	.0458	.000161	1.128	.001946	13.62	12.04	12.08	4.340	2.103	6.443
20	0.1027	0.0504	0.000177	1.237	0.002144	15.01	12.09	12.13	4.823	2.314	7.137
22	.1130	.0555	.000194	1.356	.002360	16.52	12.14	12.19	5.305	2.545	7.850
24	.1242	.0610	.000212	1.485	.002596	18.17	12.19	12.24	5.787	2.796	8.583
26	.1365	.0670	.000232	1.625	.002854	19.98	12.24	12.30	6.270	3.071	9.341
28	.1499	.0736	.000254	1.776	.003134	21.94	12.29	12.35	6.752	3.370	10.122
30	0.1646	0.0809	0.000278	1.943	0.003444	24.11	12.34	12.41	7.234	3.699	10.933
32	.1806	.0887	.000303	2.124	.003782	26.47	12.39	12.47	7.716	4.058	11.783
33	.1880	.0923	.000315	2.206	.003938	27.57	12.41	12.49	7.96	4.22	12.18
34	.1957	.0961	.000327	2.292	.004100	28.70	12.44	12.52	8.20	4.40	12.60
35	0.2036	0.1000	0.000340	2.380	0.004268	29.88	12.47	12.55	8.44	4.57	13.02
36	.2119	.1041	.000353	2.471	.004442	31.09	12.49	12.58	8.68	4.76	13.44
37	.2204	.1083	.000367	2.566	.004622	32.35	12.52	12.61	8.93	4.95	13.87
38	.2292	.1126	.000381	2.663	.004809	33.66	12.54	12.64	9.17	5.14	14.31
39	.2384	.1171	.000395	2.764	.005002	35.01	12.57	12.67	9.41	5.35	14.76
40	0.2478	0.1217	0.000410	2.868	0.005202	36.41	12.59	12.70	9.65	5.56	15.21
41	.2576	.1266	.000425	2.976	.005410	37.87	12.62	12.73	9.89	5.78	15.67
42	.2678	.1315	.000441	3.087	.005625	39.38	12.64	12.76	10.14	6.01	16.14
43	.2783	.1367	.000457	3.201	.005848	40.93	12.67	12.79	10.38	6.24	16.62
44	.2891	.1420	.000474	3.319	.006078	42.55	12.69	12.82	10.62	6.48	17.10

[a]Based on *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Below 32 F the pressure of saturated vapor in contact with ice is given.
[c]Values in this column do not include the heat of the liquid.

TABLE 2. MIXTURES OF AIR AND SATURATED WATER VAPOR[a] (CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[b]Heat Content in Btu of 1 lb of Dry Air with Vapor to Saturate it
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
45	0.3003	0.1475	0.000492	3.442	0.00632	44.21	12.72	12.85	10.86	6.73	17.59
46	.3120	.1532	.000510	3.568	.00656	45.94	12.74	12.88	11.10	6.99	18.09
47	.3240	.1591	.000528	3.698	.00682	47.73	12.77	12.91	11.34	7.26	18.60
48	.3364	.1652	.000547	3.832	.00708	49.58	12.79	12.94	11.58	7.54	19.12
49	.3492	.1715	.000567	3.970	.00736	51.49	12.82	12.97	11.83	7.83	19.65
50	0.3624	0.1780	0.000588	4.113	0.00764	53.47	12.84	13.00	12.07	8.12	20.19
51	.3761	.1848	.000609	4.260	.00793	55.52	12.87	13.03	12.31	8.43	20.74
52	.3903	.1917	.000630	4.411	.00823	57.64	12.89	13.07	12.55	8.75	21.30
53	.4049	.1989	.000653	4.568	.00855	59.83	12.92	13.10	12.79	9.08	21.87
54	.4200	.2063	.000676	4.729	.00887	62.09	12.95	13.13	13.03	9.41	22.45
55	0.4356	0.2140	0.000699	4.895	0.00920	64.43	12.97	13.16	13.28	9.76	23.04
56	.4517	.2219	.000724	5.066	.00955	66.85	13.00	13.20	13.52	10.13	23.64
57	.4684	.2300	.000749	5.242	.00991	69.35	13.02	13.23	13.76	10.50	24.25
58	.4855	.2384	.000775	5.424	.01028	71.93	13.05	13.26	14.00	10.89	24.88
59	.5032	.2471	.000802	5.611	.01066	74.60	13.07	13.30	14.24	11.28	25.52
60	0.5214	0.2561	0.000829	5.804	0.01105	77.3	13.10	13.33	14.48	11.69	26.18
61	.5403	.2654	.000858	6.003	.01146	80.2	13.12	13.36	14.72	12.12	26.84
62	.5597	.2749	.000887	6.208	.01188	83.2	13.15	13.40	14.97	12.56	27.52
63	.5798	.2848	.000917	6.418	.01231	86.2	13.17	13.43	15.21	13.01	28.22
64	.6005	.2949	.000948	6.633	.01276	89.3	13.20	13.47	15.45	13.48	28.93
65	0.6218	0.3054	0.000979	6.855	0.01323	92.6	13.22	13.50	15.69	13.96	29.65
66	.6438	.3162	.001012	7.084	.01370	95.9	13.25	13.54	15.93	14.46	30.39
67	.6664	.3273	.001046	7.320	.01420	99.4	13.27	13.58	16.18	14.97	31.15
68	.6898	.3388	.001080	7.563	.01471	103.0	13.30	13.61	16.42	15.50	31.92
69	.7139	.3506	.001116	7.813	.01524	106.6	13.32	13.65	16.66	16.05	32.71
70	0.7386	0.3628	0.001153	8.069	0.01578	110.5	13.35	13.69	16.90	16.61	33.51
71	.7642	.3754	.001190	8.332	.01634	114.4	13.38	13.73	17.14	17.19	34.33
72	.7906	.3883	.001229	8.603	.01692	118.4	13.40	13.76	17.38	17.79	35.17
73	.8177	.4016	.001269	8.882	.01751	122.6	13.43	13.80	17.63	18.41	36.03
74	.8456	.4153	.001310	9.168	.01813	126.9	13.45	13.84	17.87	19.05	36.91

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

Table 2. Mixtures of Air and Saturated Water Vapor[a] (Continued)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[b]Heat Content in Btu of 1 lb of Dry Air with Vapor to Saturate it
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
75	0.8744	0.4295	0.001352	9.46	0.01877	131.4	13.48	13.88	18.11	19.71	37.81
76	.9040	.4440	.001395	9.76	.01942	135.9	13.50	13.92	18.35	20.38	38.73
77	.9345	.4590	.001439	10.07	.02010	140.7	13.53	13.96	18.59	21.08	39.67
78	.9658	.4744	.001485	10.39	.02080	145.6	13.55	14.00	18.84	21.80	40.64
79	.9981	.4903	.001532	10.72	.02152	150.6	13.58	14.05	19.08	22.55	41.63
80	1.0314	0.5066	0.001580	11.06	0.02226	155.8	13.60	14.09	19.32	23.31	42.64
81	1.0656	.5234	.001629	11.40	.02303	161.2	13.63	14.13	19.56	24.11	43.67
82	1.1008	.5406	.001680	11.76	.02381	166.7	13.65	14.17	19.80	24.92	44.72
83	1.1370	.5584	.001732	12.12	.02463	172.4	13.68	14.22	20.04	25.76	45.80
84	1.174	.5767	.001786	12.50	.02547	178.3	13.70	14.26	20.29	26.62	46.91
85	1.212	0.5955	0.001841	12.89	0.02634	184.4	13.73	14.31	20.53	27.51	48.04
86	1.251	.6148	.001897	13.28	.02723	190.6	13.75	14.35	20.77	28.43	49.20
87	1.292	.6347	.001955	13.68	.02815	197.0	13.78	14.40	21.01	29.38	50.39
88	1.334	.6551	.002014	14.10	.02910	203.7	13.80	14.45	21.25	30.35	51.61
89	1.377	.6761	.002075	14.53	.03008	210.6	13.83	14.50	21.50	31.36	52.86
90	1.421	0.6977	0.002137	14.96	0.03109	217.6	13.86	14.55	21.74	32.39	54.13
91	1.466	.7200	.002201	15.41	.03213	224.9	13.88	14.60	21.98	33.46	55.44
92	1.512	.7427	.002267	15.87	.03320	232.4	13.91	14.65	22.22	34.59	56.78
93	1.560	.7660	.002334	16.34	.03430	240.1	13.93	14.70	22.46	35.69	58.15
94	1.609	.7901	.002403	16.82	.03544	247.1	13.96	14.75	22.71	36.86	59.56
95	1.659	0.8148	0.002474	17.32	0.03662	256.3	13.98	14.80	22.95	38.06	61.01
96	1.710	.8401	.002546	17.82	.03783	264.8	14.01	14.86	23.19	39.30	62.48
97	1.763	.8662	.002621	18.35	.03908	273.6	14.03	14.91	23.43	40.57	64.00
98	1.818	.8929	.002697	18.88	.04036	282.5	14.06	14.97	23.67	41.88	65.55
99	1.874	.9204	.002775	19.42	.04169	291.8	14.08	15.02	23.91	43.24	67.15
100	1.931	0.9486	0.002855	19.98	0.04305	301.3	14.11	15.08	24.16	44.63	68.79
101	1.990	0.9775	.002937	20.56	.04446	311.2	14.14	15.14	24.40	46.07	70.47
102	2.051	1.0072	.003021	21.15	.04591	321.4	14.16	15.20	24.64	47.54	72.18
103	2.113	1.0376	.003107	21.75	.04741	331.9	14.19	15.26	24.88	49.07	73.95
104	2.176	1.0689	.003195	22.36	.04895	342.7	14.21	15.33	25.13	50.64	75.77

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

TABLE 2. MIXTURES OF AIR AND SATURATED WATER VAPOR[a] (CONTINUED)

Temp., F	Pressure of Saturated Vapor		Weight of Saturated Vapor				Volume in Cu Ft		Heat Content in Btu of 1 lb of Dry Air Above 0 F	Latent Heat of Vapor, Btu	[b]Heat Content in Btu of 1 lb of Dry Air with Vapor to Saturate it
	In. of Hg	Lb per Sq In.	per Cu Ft		per lb of Dry Air		of 1 lb of Dry Air	of 1 lb of Dry Air + Vapor to Saturate it			
			Pounds	Grains	Pounds	Grains					
105	2.241	1.1010	0.003285	22.99	0.0505	354	14.24	15.39	25.37	52.26	77.63
106	2.308	1.134	.003377	23.64	.0522	365	14.26	15.46	25.61	53.92	79.53
107	2.377	1.168	.003472	24.30	.0539	377	14.29	15.52	25.85	55.64	81.49
108	2.448	1.202	.003568	24.98	.0556	389	14.31	15.59	26.09	57.41	83.50
109	2.520	1.238	.003667	25.67	.0574	402	14.34	15.66	26.33	59.23	85.57
110	2.594	1.274	0.003769	26.38	0.0593	415	14.36	15.73	26.58	61.11	87.69
111	2.670	1.311	.003873	27.11	.0612	428	14.39	15.80	26.82	63.04	89.86
112	2.748	1.350	.003979	27.85	.0631	442	14.41	15.87	27.06	65.04	92.10
113	2.827	1.389	.004087	28.61	.0652	456	14.44	15.95	27.30	67.10	94.40
114	2.909	1.429	.004198	29.39	.0673	471	14.46	16.02	27.55	69.22	96.77
115	2.993	1.470	0.004312	30.18	0.0694	486	14.49	16.10	27.79	71.40	99.10
116	3.079	1.512	.004428	31.00	.0717	502	14.52	16.18	28.03	73.65	101.68
117	3.167	1.555	.004547	31.83	.0739	518	14.54	16.26	28.27	75.97	104.24
118	3.257	1.600	.004669	32.68	.0763	534	14.57	16.35	28.51	78.36	106.87
119	3.349	1.645	.004793	33.55	.0788	551	14.59	16.43	28.76	80.80	109.56
120	3.444	1.692	0.004920	34.44	0.0813	569	14.62	16.52	29.00	83.37	112.37
125	3.952	1.941	.005599	39.19	.0953	667	14.75	16.99	30.21	97.33	127.54
130	4.523	2.221	.006356	44.49	.1114	780	14.88	17.53	31.42	113.64	145.06
135	5.163	2.536	.007197	50.38	.1305	913	15.00	18.13	32.63	132.71	165.34
140	5.878	2.887	.008130	56.91	.1532	1072	15.13	18.84	33.85	155.37	189.22
145	6.677	3.280	0.00916	64.1	0.1800	1260	15.26	19.64	35.06	182.05	217.1
150	7.566	3.716	.01030	72.1	.2122	1485	15.39	20.60	36.27	214.03	250.3
155	8.554	4.201	.01156	80.9	.2511	1758	15.52	21.73	37.48	252.61	290.1
160	9.649	4.739	.01294	90.6	.2987	2091	15.64	23.09	38.69	299.55	338.2
165	10.860	5.334	.01445	101.1	.3577	2504	15.77	24.75	39.91	357.75	397.7
170	12.20	5.990	0.01611	112.8	0.4324	....	15.90	26.84	41.12	431.2	472.3
175	13.67	6.71	.01793	125.5	.5290	....	16.03	29.51	42.33	526.0	568.3
180	15.29	7.51	.01991	139.4	.6577	....	16.16	33.04	43.55	651.9	695.5
185	17.07	8.38	.02206	154.4	.8359	....	16.28	37.89	44.76	826.1	870.9
190	19.01	9.34	.02441	170.9	1.0985	....	16.41	45.00	45.97	1082.3	1128.3
200	23.46	11.53	0.02972	208.0	2.2953	....	16.67	77.24	48.40	2247.5	2296

[a]Reprinted by permission from *Properties of Steam and Ammonia*, by the late G. A. Goodenough.
[b]Values in this column do not include the heat of the liquid.

the air and water. As the air passes over the water surface, it will gradually pick up water vapor and will approach saturation *at the initial wet-bulb temperature of the air*, if the water be supplied at this wet-bulb temperature. During the process of adiabatic saturation, then, the dry-bulb temperature of the air drops to the wet-bulb temperature as a limit, the wet-bulb temperature remains substantially constant, and the weight of water vapor associated with each pound of dry air increases to $W_{t'}$, as a limit, *where* $W_{t'}$ is the weight of saturated vapor per pound of dry air for saturation at the wet-bulb temperature.

Example 3. If air with a dry-bulb of 85 F and a wet-bulb of 70 F be saturated adiabatically by spraying with recirculated water, what will be the final temperature and the vapor content of the air?

Solution. The final temperature will be equal to the initial wet-bulb temperature or 70 F, and since the air is saturated at this temperature, from Table 2, W = 0.01578 lb per pound of dry air.

In the adiabatic saturation process, since the heat given up by the dry air and associated vapor in cooling to the wet-bulb temperature is utilized in evaporation of water at the wet-bulb temperature, W. H. Carrier has pointed out[2] that the equation for the process of *adiabatic saturation*, and hence for a process of *constant wet-bulb temperature*, is:

$$h'_{fg}\,(W_{t'} - W) = c_{p_a}\,(t - t') + c_{p_s} W\,(t - t') \qquad (8a)$$

and using c_{p_a} = 0.24 and c_{p_s} = 0.45

$$h'_{fg}\,(W_{t'} - W) = (0.24 + 0.45W)\,(t - t') \qquad (8b)$$

where

h'_{fg} = latent heat of vaporization at t', Btu per pound.

$(W_{t'} - W)$ = increase in vapor associated with 1 lb of dry air when it is saturated adiabatically from an initial dry-bulb temperature, t, and an initial vapor content, W, in pounds.

$0.24 + 0.45W$ = humid specific heat, in Btu per pound of dry air per degree Fahrenheit.

Knowing any two of the three primary variables, t, t', or W, the third may be found from this equation for any process of adiabatic saturation.

TOTAL HEAT AND HEAT CONTENT

The total heat of a mixture of dry air and water vapor was originally defined by W. H. Carrier as follows:

$$\Sigma = c_{p_a}\,(t - 0) + W\,[h'_{fg} + c_{p_s}\,(t - t')] \qquad (9)$$

where

Σ = total heat of the mixture in Btu per pound of dry air.
c_{p_a} = mean specific heat at constant pressure of dry air.
c_{p_s} = mean specific heat at constant pressure of water vapor.
t = dry-bulb temperature, in degrees Fahrenheit.
t' = wet-bulb temperature, in degrees Fahrenheit.
W = weight of water vapor mixed with each pound of dry air, in pounds.
h'_{fg} = latent heat of vaporization at t', in Btu per pound.

[2] *A.S.M.E. Transactions*, Vol. 33, 1911, p. 1005.

Since this definition holds for any mixture of dry air and water vapor, the total heat of a mixture with a relative humidity of 100 per cent and at a temperature equal to the wet-bulb temperature (t') is

$$\Sigma' = c_{p_a}\,(t' - 0) + W_{t'}\,h'_{fg} \tag{10}$$

By equating Equation 9 to Equation 10, the equation for the adiabatic saturation process, Equation 8a, follows. This demonstrates that the adiabatic saturation process at constant wet-bulb temperature is also a process of constant total heat. In short, the total heat of a mixture of dry air and water vapor is the same for any two states of the mixture at the same wet-bulb temperature. This fact furnishes a convenient means of finding the total heat of an air-vapor mixture in any state.

Example 4. Find the total heat of an air-vapor mixture having a dry-bulb temperature of 85 F and a wet-bulb temperature of 70 F.

Solution. From Table 2, for saturation at the wet-bulb temperature, $W_{t'} = 0.01578$, and from Equation 10,

$$\Sigma' = c_{p_a}\,(70 - 0) + 0.01578\,h'_{fg} = 16.9 + 16.61 = 33.51$$

By considering the temperatures in Table 2 to be wet-bulb readings, the total heat of any air-vapor mixture may be obtained from the last column in the table.

Enthalpy

This total heat of an air-vapor mixture is not exactly equal to the true heat content or enthalpy of the mixture since the heat content of the liquid is not included in Equation 9. With the meaning of heat content in agreement with present practise in other branches of thermodynamics, the true heat content of a mixture of dry air and water vapor (with 0 deg F as the datum for dry air and the saturated liquid at 32 F as the datum for the water vapor) is

$$h = c_{p_a}\,(t - 0) + W\,h_s = 0.24\,(t - 0) + W\,h_s \tag{11}$$

where

h = the heat content of the mixture, in Btu per pound of dry air.
t = the dry-bulb temperature, in degrees Fahrenheit.
W = the weight of vapor per pound of dry air, in pounds.
h_s = the heat content of the vapor in the mixture, in Btu per pound.

The heat content of the water vapor in the mixture may be found in steam charts or tables when the dry-bulb temperature and the partial pressure of the vapor are known. Or, since the heat content of steam at low partial pressures, whether super-heated or saturated, depends only upon temperature, the following empirical equation, derived from Keenan's Steam Tables, may be used:

$$h_s = 1059.2 + 0.45\,t \tag{12}$$

Substituting this value of h_s in Equation 11, the heat content of the mixture is

$$h = 0.24\,(t - 0) + W\,(1059.2 + 0.45\,t) \tag{13}$$

An energy equation can be written that applies, in general, to various

air-conditioning processes, and this equation can be used to determine the quantity of heat transferred during such processes. In the most general form, this equation may be explained with the aid of Fig. 1 as follows:

The rectangle may represent any apparatus, *e.g.*, a drier, humidifier, dehumidifier, cooling tower, or the like, by proper choice of the direction of the arrows.

In general, a mixture of air and water vapor, such as atmospheric air, enters the apparatus in State 1 and leaves in State 3. Water is supplied at some temperature, t_2. For the flow of 1 lb of dry air (with accompanying vapor) through the apparatus, providing there is no appreciable change in the elevation or velocity of the fluids and no mechanical energy delivered to or by the apparatus, then

$$h_1 + E_h + (W_3 - W_1)\, h_2 = h_3 + R_c$$

or

$$E_h - R_c = h_3 - h_1 - (W_3 - W_1)\, h_2 \qquad (14)$$

where

E_h = the quantity of heat supplied per pound of dry air, in Btu.

R_c = the quantity of heat lost externally by heat transfer from the apparatus, in Btu per pound of dry air.

W_1 = the weight of water vapor entering, per pound of dry air.

W_3 = the weight of water vapor leaving, per pound of dry air.

h_2 = the heat content of the water supplied at t_2, in Btu per pound.

$h_3 - h_1$ = the increase in the heat content of the air-water vapor mixture in passing through the apparatus, in Btu per pound of dry air
$= 0.24\,(t_3 - t_1) + W_3\,(1059.2 + 0.45\,t_3) - W_1\,(1059.2 + 0.45 t_1)$

The net quantity of heat added to or removed from air-water vapor mixtures in air conditioning work is frequently *approximated* by taking the differences in total heat at exit and entrance.

For example, in Fig. 1, an *approximate* result is

$$E_h - R_c = \Sigma_3 - \Sigma_1 \qquad (15)$$

where

Σ_3 = the total heat of the air-vapor mixture at exit, in Btu per pound of dry air.

Σ_1 = the total heat of the air-vapor mixture at entrance, in Btu per pound of dry air.

From the definitions of *total heat* and *heat content*, it may be demonstrated that Equation 15 is *exactly* equivalent to Equation 14, when, and only when, $t'_3 = t'_1 = t_2$; *i.e.*, when the initial and final wet-bulb temperatures and the temperature of the water supplied are equal. The one process that meets these conditions is the process of adiabatic saturation, and either equation will give a result of zero for this process; for other conditions, Equation 15 is approximate but satisfactory for many calculations.

The following problems illustrate the application of these principles:

Example 5. Heating (data from Example 1). Assuming the water to be supplied at 50 F, the net quantity of heat supplied is, from Equation 14,

$$E_h - R_c = 0.24\ (70 - 0) + 0.000547 \times 0.45\ (70 - 0) + 0.005633$$
$$[1059.2 + 0.45 \times 70 - (50 - 32)] = 22.87 \text{ Btu per pound of dry air}$$

Example 6. Cooling (data from Example 2). If the condensate is removed at 54 F the quantity of heat removed is found from Equation 14, by proper regard to the arrow direction in Fig. 1,

$$E_h + R_c = 0.24\ (84 - 54) + 0.00887 \times 0.45\ (84 - 54) + 0.00358$$
$$[1059.2 + 0.45 \times 84 - (54 - 32)] = 11.17 \text{ Btu per pound of dry air}$$

Using Table 2, the initial total heat of the air-vapor mixture, since the wet-bulb temperature is 70 F, is 33.51 Btu per pound of dry air.

The final total heat is, from Table 2, since the exit air is saturated, 22.45 Btu per pound. Hence, using Equation 15, the quantity of heat removed is, approximately, (33.51 − 22.45) or 11.06 Btu per pound of dry air. The degree of approximation to the correct result is evident in this example.

PSYCHROMETRIC CHART[3]

The Bulkeley Psychrometric Chart[4] (Fig. 2) shows graphically the relationships expressed in Equations 8a and 8b. It also gives the grains of moisture per pound of dry air for saturation, the grains of moisture per cubic foot of saturated air, the total heat in Btu per pound of dry air saturated with moisture, and the weight of the dry air in pounds per cubic foot. Fig. 3 shows the procedure to follow in using the Bulkeley Chart. The directrix curves above the saturation line are as follows:

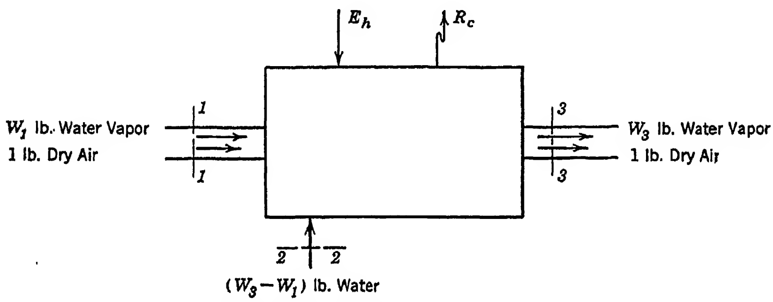

FIG. 1. DIAGRAM ILLUSTRATING ENERGY EQUATION 14

A is the total heat in Btu contained in the mixture above 0 F, and is to be referred to column of figures at left side of chart.

B is the grains of moisture of water vapor contained in each pound of the saturated mixture and is to be referred to the figures at the left side of the chart.

C is the grains of moisture of water vapor per cubic foot of saturated mixture, and is to be referred to the figures at the left side of the chart and divided by ten.

D is the weight in decimal fractions of a pound, of one cubic foot of the saturated mixture, and is referred to the first column of figures to the right of the saturation line between the vertical dry-bulb temperature lines 170 and 180 F. The relative density of the mixture is read in a similar manner from the same curve by the column of figures between the vertical dry-bulb temperature lines 180 and 190 F.

E is similar to *D* but is for dry air, devoid of all moisture or water vapor. For convenience, the approximate absolute temperature of 500 F is given at 40 F on the saturation line for the purpose of calculating volume, weight per cubic foot, and relative density at partial saturation.

Example 7. Relative Humidity: At the intersection of the 78 F wet-bulb line and the 95-F dry-bulb line, the relative humidity is read directly on the straight diagonal lines as 45.7 per cent.

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[3]See, A Review of Psychrometric Charts, C. O. Mackey (*Heating and Ventilating*, June, July, 1931).

[4]The Bulkeley Psychrometric Chart was presented to the Society in 1926. (See A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926). Fig. 2 has been reproduced from the original chart but with the secondary cross-section lines omitted.

Example 8. Dew-Point: At the intersection of the 78 F wet-bulb line, the dew-point temperature is read directly on the horizontal temperature lines as 70.9 F.

Example 9. Vapor Pressure: At the intersection of the 78 F wet-bulb line and the 95 F dry-bulb line, pass in a horizontal direction to the left of the chart and on the

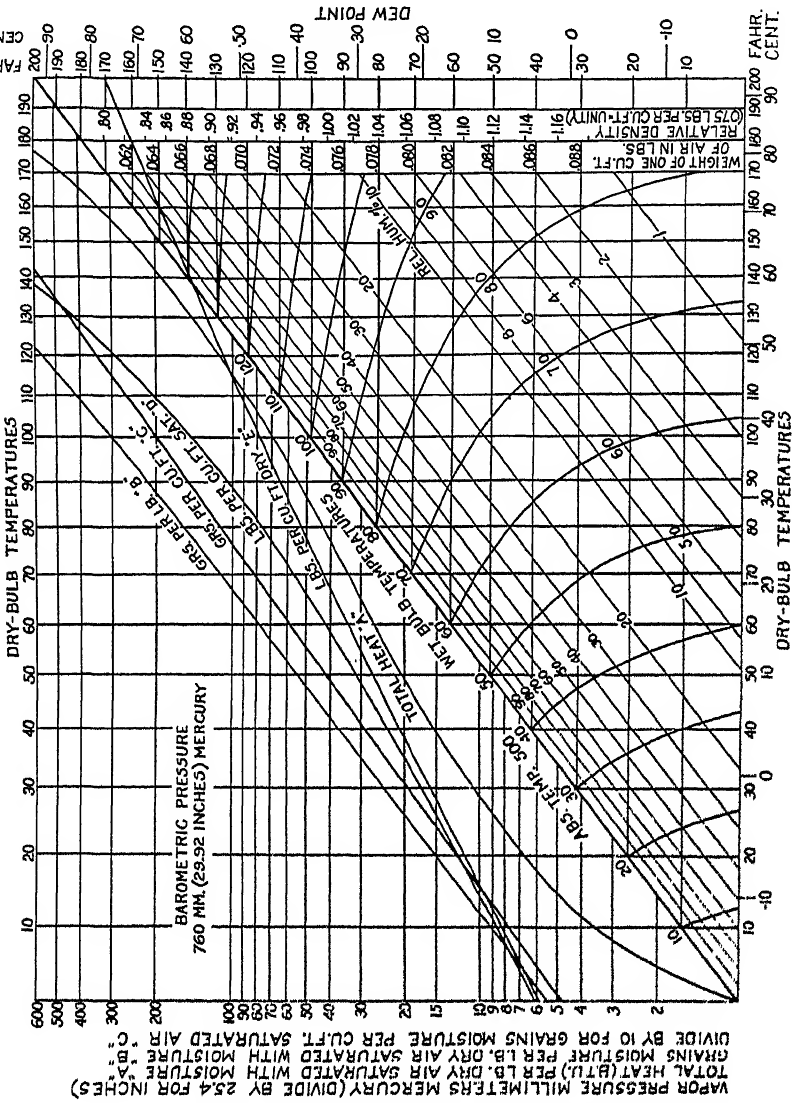

FIG. 2. PSYCHROMETRIC CHART

logarithmic scale read the vapor pressure as 19.4 millimeters of mercury. (Divide by 25.4 for inches).

Example 10. Total Heat Above 0 Deg in Mixture per Pound of Dry Air Saturated with Moisture: From where the wet-bulb line joins saturation line, pass in vertical direction on 78-F dry-bulb line to its intersection with curve *A* and on the logarithmic scale at the left of the chart read 40.6 Btu per pound of mixture. The use of this curve to obtain

total heat in the mixture at any wet-bulb temperature is a great convenience, as the number of Btu required to heat the mixture and humidify, as well as the refrigeration required to cool and dehumidify the mixture, can be obtained by taking the difference in total heat before and after treatment of the mixture.

Example 11. Grains of Moisture per Pound of Mixture: From 70.9 F dew-point temperature on the saturation line, pass vertically to the intersection with curve B and on the logarithmic scale at the left read 114 grains of moisture per pound.

Example 12. Grains of Moisture per Cubic Foot of Mixture, Partially Saturated: From 70.9 F dew-point temperature on the saturation line proceed in a vertical direction to curve C, and on the logarithmic scale to the left read 83.3, which divided by 10, gives 8.33 grains. A temperature of 70.9 F is equal to an absolute temperature of 530.9, and 95 F equals 555, absolute temperature. Therefore, $\frac{530.9}{555} \times 8.33 = 7.97$ grains per cubic foot of partially saturated mixture.

Example 13. Grains of Moisture per Cubic Foot of Dry Air, Saturated: Starting at the saturation line at the desired temperature, pass in vertical direction to curve C and on logarithmic scale, on left, read and divide by 10.

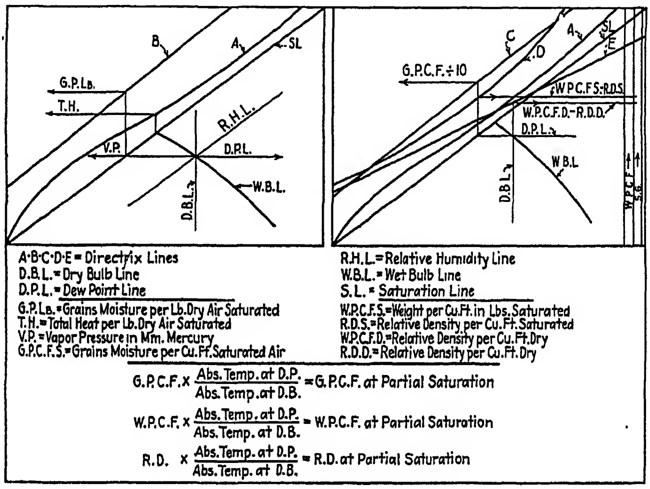

FIG. 3. DIAGRAMS SHOWING PROCEDURE TO FOLLOW IN USING BULKELEY CHART

Example 14. Weight per Cubic Foot of Dry Air and Relative Density: From point where, for example, the 70-F vertical dry-bulb line intersects curve E, pass to right side and read 0.075 lb; if cubic feet per pound is desired, divide 1 by this amount. The relative density is read immediately to the right as 1.00.

Example 15. Weight per Cubic Foot of Saturated Air and Relative Density: From point where, for example, the 70 F vertical line intersects the curve D, pass to the right and read weight per cubic foot as 0.07316 with a relative density of 0.9755 for saturated air at 70 F.

Example 16. Weight per Cubic Foot and Relative Density of Partially Saturated Air: Air at 50 F and a wet-bulb temperature of 46 F is to be heated to 130 F. The wet- and dry-bulb lines intersect at a dew-point temperature of 42 F. Pass to the left where this dew-point line intersects the saturation line and then pass in a vertical direction to where the 42 F dry-bulb line intersects with curve D. Then pass directly to the right and read the weight per cubic foot of saturated air at 42 F as 0.07844 and the relative density as 1.046. The absolute temperature at 42 F is 502, and at 130 F is 590. Therefore, $\frac{502}{590} = 0.851$. The weight of 1 cu ft of air at 50 F dry-bulb and 46 F wet-bulb when heated to 130 F is $0.07844 \times 0.851 = 0.06675$, and the relative density is $1.046 \times 0.851 = 0.89$.

RATE OF EVAPORATION

In problems of air conditioning and drying, as well as in other industrial applications of evaporation, such as cooling towers, for example, it is desirable to determine the rate of evaporation. There are two distinct cases of evaporation. The *first* case is that in which the source of heat is primarily from the water itself and in which the air temperature may even be raised. The *second* is that in which the heat for evaporation is obtained entirely from the air itself, in which case the air is cooled and the temperature of the water remains substantially constant at the wet-bulb temperature. Both cases, however, may be reduced to a common basis of calculation. It has been found that the increase in the rate of evaporation is nearly in direct proportion to the increase in the air velocity, and that it is in direct proportion to the difference in vapor pressure between the vapor pressure of the water and the pressure of the vapor in the air.

The general formula covering the experimental data may be expressed as follows:

$$\frac{dw}{dt} = (a + bv)\,(e' - e) \qquad (16)$$

where

$\frac{dw}{dt}$ = rate of evaporation.
a = the rate of evaporation in still air.
b = the rate of increase with velocity.
e' = the vapor pressure of the liquid.
e = the vapor pressure in the atmosphere.
v = velocity.

The only difference between case *one* and case *two* is that in case *one* the vapor pressure of the liquid is one of the known or assumed factors, being dependent upon the known temperature of the liquid, while in case *two*, e' is the vapor pressure corresponding to the wet-bulb temperature of the air.

This wet-bulb or evaporation temperature is dependent upon the dry-bulb temperature and the moisture content, or upon the total heat of the air as indicated in the previous paragraph.

The effect of air velocity depends upon whether the flow of air is parallel to the surface or perpendicular to the surface elements. For a flow of air *parallel* to a horizontal surface

$$w = 0.093\left(1 + \frac{v}{230}\right)(e' - e) \quad \text{(approximately)} \qquad (17)$$

where

w = pounds evaporated per square foot per hour.
v = velocity of atmosphere over surfaces in feet per minute.
e' = vapor pressure of the water corresponding to its temperature.
e = vapor pressure in the surrounding atmosphere.

For *transverse* flow, as across a tubular surface, the rate of evaporation is nearly doubled.

These relationships are indicated graphically on the chart, Fig. 4.

Since the difference in vapor pressures is substantially proportional to the difference between the wet- and dry-bulb temperatures (*i.e.*, the wet-bulb depression) the rate of evaporation is also for case *two* substantially proportionate to the wet-bulb depression.

In case *two*, the rate of sensible heat transfer from the air to the liquid to produce evaporation is substantially the same as the rate of heat transfer with the same type of surface, without moisture being present, but with the same temperature differences. In other words, the rate of heat transfer depends upon the temperature difference only, whether the

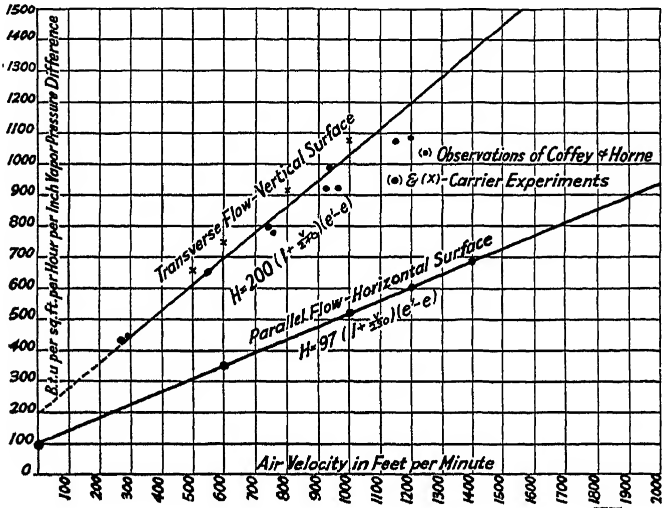

FIG. 4. HEAT TRANSMITTED BY EVAPORATION

surface is wet or not. For example, it has been shown that the rate of heat transfer with air flowing across staggered coils (transverse flow) may be represented by the formula:

$$U_t = \frac{1}{0.0447 + \dfrac{50.66}{v}} \tag{18}$$

where

U_t = heat transfer expressed in Btu per hour per square foot per degree difference in temperature between steam and air, for transverse flow.

At a velocity of 400 fpm, U_t = 5.8; at a velocity of 800 fpm, U_t = 9.3.

Referring to Fig. 4, showing the rate of heat transmission by evaporation for different air velocities, it will be noted that for transverse flow there are 560 Btu per hour per square foot transferred per inch difference of vapor pressure at a velocity of 400 fpm and 910 Btu per hour per square foot per inch difference in vapor pressure at a velocity of 800 fpm. One inch of vapor pressure difference corresponds approximately to 95 deg difference between the wet- and dry-bulb temperature. Dividing by 95,

the value of 5.9 Btu per square foot per degree difference in temperature is obtained for a velocity of 400 fpm and 9.55 Btu per square foot for a velocity of 800 fpm.

It will be noted that for these two cases the heat transfer by evaporation per degree difference in temperature corresponds almost exactly with the heat transfer by convection coils. The similarity may be noted by comparing the formula for heat transfer in parallel flow, where

$$U_{\mathrm{p}} = \frac{1}{0.026 + \frac{161}{v}} \tag{19}$$

with the heat transfer by evaporation with parallel flow. The relationship will be seen to be very close in both cases and would indicate that the heat transfer by evaporation is actually brought about by a process of convection.

The difference in form of the two formulae may be due in part to errors in observation at the higher and lower velocities.

In cooling air and condensing out the moisture therefrom the heat transfer is considerably more rapid than when the air is dry and no moisture is condensed. In general the rate of heat transmission on the air side is increased an amount which is proportionate to the latent heat removed as compared with the sensible heat removed. That is, if the latent heat removed was 50 per cent of the sensible heat removed, then the conductivity of the surface in contact with the air would be increased approximately 50 per cent.

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Chapter 2

VENTILATION AND AIR CONDITIONING STANDARDS

Vitiation of Air, Heat Regulation in Man, Effects of Heat, Effects of Cold, Temperature Changes, Acclimatization, Warmth and Comfort, Effective Temperature, Comfort Chart, Comfort Line, Comfort Zone, Application of Comfort Chart, A.S.H.V.E. Ventilation Standards, Natural and Mechanical Ventilation, Recirculation, Ozone, Ultra-Violet Radiation and Ionization, Heat and Moisture Losses

VENTILATION is defined in part as "the process of supplying or removing air by natural or mechanical means to or from any space." (See Chapter 42). The word in itself implies quantity but not necessarily quality. From the standpoint of comfort and health, however, the problem is now considered to be one of securing air of the proper quality rather than of supplying a given quantity.

The term *air conditioning* implies both air quality and quantity. In addition to air change, its scope is to control simultaneously the temperature, humidity, air movement and purity. The term is broad enough to embrace whatever other additional factors may be found desirable for maintaining the atmosphere of occupied spaces at a condition best suited to the physiological requirements of the human body.

VITIATION OF AIR

Under the artificial conditions of indoor life, the air undergoes certain chemical changes and a vitiation which are brought about by the occupants themselves. The oxygen content is somewhat reduced, and the carbon dioxide slightly increased by the respiratory processes. Organic matter, which is usually perceived as odors, comes from the nose, mouth, skin and clothing. The temperature of the air is increased by the metabolic processes, and the humidity raised by the moisture emitted from the skin and lungs. Moreover, according to latest researches[1], there is a marked decrease in both positive and negative ions in the air of occupied rooms.

Contrary to old theories, the usual changes in oxygen and carbon dioxide are of no physiological concern because they are much too small even under the worst conditions. The amount of carbon dioxide in air is often used in ventilation work as an index of odors of human origin, but the

[1]See A.S.H.V.E. research paper entitled, Changes in Ionic Content in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

information it affords rarely justifies the labor involved in making the observation[2]. Little is known of the identity and physiological effects of the organic matter given off in the process of respiration. The former belief that the discomfort experienced in confined spaces was due to some toxic volatile matter in the expired air is now limited, in the light of numerous researches, to the much less dogmatic view that the presence of such a substance has not been demonstrated. The only fact that does appear certain is that expired and transpired air is odorous and offensive, and it is capable of producing headache, nausea, loss of appetite and a disinclination for physical activity. These reasons alone, whether æsthetic or physiological, are sufficient to warrant proper air conditions.

A certain part of the dissemination of disease which occurs in confined spaces is caused by the continuous emission of pathogenic bacteria from infected persons. Infections by droplets from coughing and sneezing constitute a limited mode of transmission in the immediate vicinity of the infected person. Experiments have shown that the mouth spray is a coarse rain which settles down quickly. The contamination is local and the problem is considered to be largely one of contact infection rather than air-borne infection.

The primary factors in air conditioning work, in the absence of any specific contaminating source, are temperature, humidity, air movement and body odors. As compared with these physical factors, the chemical factors are, as a general rule, of secondary importance.

HEAT REGULATION IN MAN

The importance of temperature, humidity and air movement arises from the profound influence which these factors exert upon body temperature, comfort and health. Body temperature is a resultant of the balancing action between its heat production and its heat loss. The heat resulting from the combustion of food within the body maintains its temperature well above that of the surrounding air. At the same time, heat is constantly lost from the body by radiation, conduction and evaporation. Since, under ordinary conditions, the body temperature is maintained at its normal level of about 98.6 F, the heat production must be balanced by the heat loss. In healthy persons this takes place automatically by the action of the heat regulating mechanism.

According to the general view, special areas in the skin are sensitive to temperature. Nerve courses carry the sense impressions to the brain and the response comes back over another set of nerves, the motor nerves, to the musculature and to all the active tissues in the body, including the endocrine glands. In this way, a two-sided mechanism controls the body temperature by (1) regulation of internal heat production (chemical regulation), and (2) regulation of heat loss by means of automatic variation in the rate of cutaneous circulation and the operation of the sweat glands (physical regulation). The mechanisms of adjustment are complex and little understood at the present time. Coordination of these different mechanisms seems to vary greatly with different air conditions.

[2]Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933, p. 324).

TABLE 1. PHYSIOLOGICAL RESPONSES TO HEAT OF MEN AT REST AND AT WORK[a]

Effective Temp.	Actual Cheek Temp (Deg Fahr)	Men at Rest			Men at Work 90,000 ft-lb of Work per Hour			
		Rise in Rectal Temp (Deg Fahr per Hour)	Increase in Pulse Rate (Beats per Min per Hour)	Approximate Loss in Body Weight by Perspiration (lb per Hr)	Total Work Accomplished (ft-lb)	Rise in Body Temp (Deg Fahr per Hr)	Increase in Pulse Rate (Beats per Min per Hr)	Approximate Loss in Body Wt. by Perspiration (lb per Hr)
60	....	....	....	....	225,000	0.0	6	0.5
70	....	0.0	0	0.2	225,000	0.1	7	0.6
80	96.1	0.0	0	0.3	209,000	0.3	11	0.8
85	96.6	0.1	1	0.4	190,000	0.6	17	1.1
90	97.0	0.3	4	0.5	153,000	1.2	31	1.5
95	97.6	0.9	15	0.9	102,000	2.3	61	2.0
100	99.6	2.2	40	1.7	67,000	4.0	103[b]	2.7
105	104.7	4.0	83	2.7	49,000	6.0[b]	158[b]	3.5[b]
110	....	5.9[b]	137[b]	4.0[b]	37,000	8.5[b]	237[b]	4.4[b]

[a]Data by A.S.H.V.E. Research Laboratory.
[b]Computed value from exposures lasting less than one hour.

In reasonably warm environments (75 F to 80 F), metabolism, or internal heat production, is decreased to some extent, probably by an inhibitory action on heat producing organs, such as the liver. The blood capillaries in the skin become dilated by reflex action of the vasomotor nerves, allowing more blood to flow into the skin, and thus increase its temperature and consequently its heat loss. The increase in peripheral circulation is at the expense of the internal organs. If this method of cooling is not in itself sufficient, the stimulus is extended to the sweat glands which allow water to pass through the surface of the skin, where it is evaporated. This method of cooling is the most effective of all, as long as the humidity of the air is sufficiently low to allow for evaporation. In high humidities, equally good results may be obtained by increasing the air movement, and hence the heat loss by conduction and evaporation.

In cold environments, in order to keep the body warm there is an actual increase in metabolism brought about partly by voluntary muscular contractions (shivering) and partly by an involuntary reflex upon the heat producing organs. The surface blood vessels become constricted and shrink farther below the surface, thus increasing the insulating layer and decreasing heat loss. The blood supply to the skin is curtailed by vasomotor shifts to the internal organs, in order to conserve body heat.

EFFECTS OF HEAT

Although the human organism is capable of adapting itself to variations in environmental conditions, its ability to maintain heat equilibrium is limited. The heat regulating center fails, for instance, if the external temperature is so abnormally high that bodily heat cannot be eliminated as fast as it is produced. Part of it is retained in the body, causing a rise in skin and deep tissue temperature, an increase in the heart rate, and accelerated respiration. (See Table 1). In extreme conditions, the metabolic rate is markedly increased owing to the excessive rise in body

temperature, and a vicious cycle results which may eventually lead to serious physiologic damage.

Examples of this are met with in unusually hot summer weather and in hot industries where the radiant heat from hot objects renders heat loss from the body by radiation and convection impossible. Consequently, the workers depend entirely on evaporation for the elimination of body heat. They stream with perspiration and drink liquids abundantly to replace the loss.

One of the most deleterious effects of high temperatures is that the blood is diverted from the internal organs to the surface capillaries, in order to serve in the process of cooling. This affects the stomach, heart, lungs and other vital organs, and it is believed that the feeling of lassitude and discomfort experienced is due to the anæmic condition of the brain. The stomach loses some of its power to act upon the food, owing to a diminished secretion of gastric juice, and there is a corresponding loss in the antiseptic and antifermentive action which favors the growth of bacteria in the intestinal tract[3]. These are considered to be the potent factors in the increased susceptibility to gastro-intestinal disorders in hot summer weather. The victim may lose appetite and suffer from indigestion, headache and general enervation, which may eventually lead to a premature old age.

In warm atmospheres, particularly during physical work, a considerable amount of chloride is lost from the system through sweating. The loss of this substance may lead to attacks of cramps, unless the salts are replaced in the drinking water. In order to relieve both cramps and fatigue, Moss[4] recommends the addition of 6 grams of sodium chloride and 4 grams of potassium chloride in a gallon of water.

The deleterious physiologic effects of high temperatures exert a powerful influence upon physical activity, accidents, sickness and mortality. Both laboratory and field data show clearly that physical work in warm atmospheres is a great effort, and that production falls progressively as the temperature rises. The incidence of industrial accidents reaches a minimum at about 68 F, increasing above and below that temperature. Sickness and mortality rates increase progressively as the temperature rises.

EFFECTS OF COLD

The action of cold on human beings is not well known. Cold affects the human organism in two ways: (1) through its action on the body as a whole, and (2) through its action on the mucous membranes of the upper respiratory tract. Little exact information is available on the latter.

On exposure to cold, the loss of heat is increased considerably and only within certain limits is compensation possible by increased heat production and decreased peripheral circulation. The rectal temperature often rises upon exposure to cold but the pulse rate and skin temperature fall.

[3]Influence of Effective Temperature upon Bactericidal Action of Gasto-Intestinal Tract, by Arnold and Brody (*Proceedings Society* Exp. Biol. Med. Vol. 24, 1927, p. 832).

[4]Some Effects of High Air Temperatures upon the Miner, by K. N. Moss (*Transactions Institute of Mining Engineers*, Vol. 66, 1924, p. 284).

The blood pressure increases, owing to constriction in the peripheral vessels and to thickening of the blood. The skin subcutaneous tissues and muscles form reservoirs for storing the water which leaves the blood. In extremely cold atmospheres compensation becomes inadequate. The body temperature falls and the reflex irritability of the spinal cord is markedly affected. The organism may finally pass into an unconscious state which ends in death.

Cannon showed that excessive loss of heat is associated with increased activity of the adrenal medulla[5]. The extra output of adrenin hastens heat production which protects the organism against cooling. Bast[6] found a degeneration of thyroid and adrenal glands upon exposure to cold, and some physicians believe that the common use of tonics in spring has something to do with the degeneration of the endocrine glands.

Effects of Temperature Changes

A moderate amount of variability in temperature is known to be beneficial to health, comfort, and the performance of physical and mental work. On the other hand, extreme changes in temperature, such as those experienced in passing from a warm room to the cold air out of doors, appear to be harmful to the tissues of the nose and throat which are the portals for the entry of respiratory diseases.

Experiments show that chilling causes a constriction of the blood vessels of the palate, tonsils and throat, which is accompanied by a fall in the temperature of the tissues. On rewarming, the palate and throat do not always regain their normal temperature and blood supply. This anæmic condition favors bacterial activity and it is believed to play a part in the inception of the common cold and other respiratory diseases. It is believed that the lowered resistance is due to a diminution in the number and phagocytic activity of the leucocytes (white blood cells) brought about by exposure to cold and by changes in temperature.

Sickness records in industries seem to strengthen this belief. The Industrial Fatigue Research Board of England[7] found that in workers exposed to high temperatures and to changes in temperature, namely, steel melters, puddlers, and tin-plate millmen, there is an excess of all sickness, the excess among the puddlers being due chiefly to respiratory diseases and rheumatism. The causative factor was not the heat itself but the sudden changes in temperature to which the workers were exposed. The tin-plate millmen who were not exposed to chills, since they work almost continuously throughout the shift, had no excess of rheumatism and respiratory diseases. On the other hand, the blast-furnacemen, who work mostly in the open, showed more respiratory sickness than the steel workers. This experience in British factories is well in accord with the findings in American industries[8]. According to these data the highest

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[5]Studies on the Condition of Activity of Endocrine Glands, by W. B. Cannon, A. Guerido, S. W. Britton and E. M. Bright (*American Journal of Physiology*, Vol. 79, 1926, p. 466).

[6]Studies in Exhaustion Due to Lack of Sleep, by T. H. Bast, J. S. Supernaw, B. Lieberman and J. Munro (*American Journal of Physiology*, Vol. 85, 1928, p. 135).

[7]Fatigue and Efficiency in the Iron and Steel Industry, by H. M. Vernon (*Industrial Fatigue Research Board*, Report No. 5, 1920, London).

[8]Iron Foundry Workers Show Highest Percentage of Deaths from Pneumonia (*Statistical Bulletin*, Metropolitan Life Insurance Company, 1928).

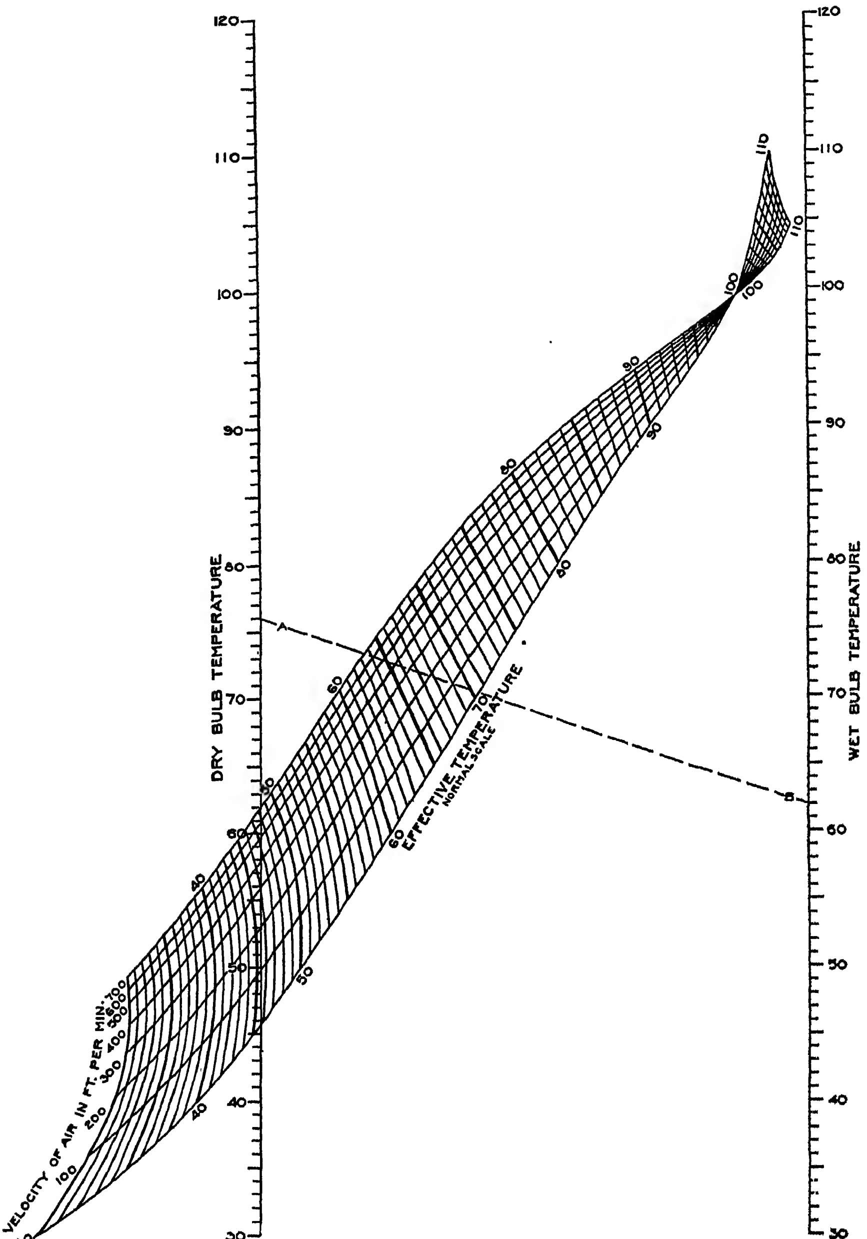

FIG. 1. THERMOMETRIC OR EFFECTIVE TEMPERATURE CHART SHOWING NORMAL SCALE OF EFFECTIVE TEMPERATURE. APPLICABLE TO INHABITANTS OF THE UNITED STATES UNDER FOLLOWING CONDITIONS:

A. Clothing: Customary indoor clothing. *B. Activity:* Sedentary or light muscular work. *C. Heating Methods:* Convection type, *i.e.* warm air, direct steam or hot water radiators, plenum systems.

pneumonia death rate is associated with dust, extreme heat, exposure to cold, and to sudden changes in temperature.

ACCLIMATIZATION

Acclimatization and the factor of psychology are two important influences in air conditioning which cannot be ignored. The first is man's ability to adapt himself to changes in air conditions; the second is an intangible matter of habit and suggestion.

Some persons regard the unnecessary endurance of cold as a virtue. They believe that the human organism can adapt itself to a wide range of air conditions with no apparent discomfort or injury to health. In the light of the present knowledge of air conditioning these views are not justified. Acclimatization to extreme conditions involves a strain upon the heat regulating system and it interferes with the normal physiologic functions of the human body. Thousands of years in the heat of Africa do not seem to have acclimatized the Negro to a temperature averaging 80 F. The same holds true of northern races with respect to cold, although the effects are mitigated by artificial control. All this seems to indicate that adaptation to a climate averaging between 60 and 80 F is a very primitive trait[9].

In certain individuals the psychologic factor is more powerful than acclimatization. A fresh air fiend may suffer in a room with windows closed regardless of the quality of the air. As a matter of fact, instances are known in which paid subjects refused to stay in a windowless but properly conditioned experimental chamber because the atmosphere felt suffocating to them upon entering the room.

SENSATIONS OF WARMTH AND COMFORT

Temperature, humidity and air motion taken together determine the feeling of warmth and influence the elimination of body heat. In other words, the temperature sensations of the human body depend not only on the temperature of the surrounding air as registered by a dry-bulb thermometer, but also upon the temperature indicated by a wet-bulb thermometer. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content. Air motion makes any moderate condition feel cooler.

On the other hand, in cold environments an increase in humidity produces a cooler sensation. The dividing line at which humidity has no effect upon comfort varies with the air velocity and is about 46 F (dry-bulb) for still air and about 51, 56 and 59 F for air velocities of 100, 300 and 500 fpm, respectively.

Thermo-Equivalent Conditions

Combinations of temperature, humidity and air movement which produce the same feeling of warmth are called thermo-equivalent conditions. Elaborate experiments made by the A.S.H.V.E. Research Laboratory

[9]Civilization and Climate, by Ellsworth Huntington, Yale University Press, 1924.

show that this newly-developed scale of thermo-equivalent conditions not only indicates the sensation of warmth, but also determines the physiological *effects* on the body induced by heat and cold. For this reason, it is called the *Effective Temperature* scale or index.

Effective temperature is an index of warmth or cold. It is not in itself an index of comfort, as it is often assumed to be, nor are the effective temperature lines necessarily lines of equal comfort. This is true because, in determining this index, the subjects compared not the relative comfort, but rather the relative warmth or cold of various air conditions. Moist air at a comparatively low temperature, and dry air at a higher temperature may each feel as warm as air of an intermediate temperature and humidity, but the *comfort* experienced in the three air conditions would be quite different, although the effective temperature is the same. The intermediate condition may be entirely comfortable, but the other two would not necessarily be so.

Under extreme humidity conditions there seems to be a difference between sensations of absolute comfort and of the proper degree of warmth. In other words, human beings are not necessarily comfortable when the air is neither *too warm* nor *too cold.* Air of proper warmth may, for instance, contain excessive water vapor, and in this way interfere with the normal physiologic loss of moisture from the skin, leading to damp skin and clothing and producing more or less discomfort; or the air may be excessively dry, producing appreciable discomfort to the mucous membrane of the nose and to the skin which dries up and becomes chapped from too rapid loss of moisture. According to the comfort experiments first conducted at the A.S.H.V.E. Laboratory in the U. S. Bureau of Mines, Pittsburgh, and later studies at the Harvard School of Public Health in Boston, effective temperature appears to be a fair index of comfort also, but only within a humidity range of 30 to 60 per cent, approximately.

Definition of Effective Temperature

Briefly, *effective temperature* may be defined as an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is not a temperature at all; it is a composite index which combines the readings of temperature, humidity and air motion in a single value. The numerical value of the effective temperature index for any given air condition is fixed by the temperature of saturated air which, at a velocity or turbulence of 15 to 25 fpm, induces a sensation of warmth or cold like that of the given condition. Thus, an air condition has an effective temperature of 65 deg when it induces a sensation of warmth like that experienced in practically still air at 65 F saturated with moisture.

In all reports of the A.S.H.V.E. Research Laboratory, the term *still air* signifies the minimum air movement it was possible to obtain in the Laboratory's psychrometric chamber. Actually, the air motion was between 15 and 25 fpm in all experiments, without qualification, as measured by the Kata thermometer. This was not a linear movement of air but it represented the turbulence or eddy currents produced by the air change. Even in tightly sealed rooms, the natural air movement is not

likely to fall below 10 fpm so long as there is a temperature or pressure difference between the air inside and outside the room.

A series of tests has been carried out in the psychrometric rooms of the A.S.H.V.E. Research Laboratory, Pittsburgh, in order to determine the equivalent conditions met with in general air conditioning work. Reports of these studies for both still and moving air are given in A.S.H.V.E. TRANSACTIONS, Vols. 27 to 38, inclusive. Fig. 1 shows the results in a single chart, the so-called thermometric chart. The equivalent conditions or effective temperature lines are shown by the short cross-lines. The difference between the effective temperature for still air and for moving air, of any velocity, represents the cooling resulting from that air velocity.

The thermometric chart (Fig. 1) applies to average normal and healthy persons adapted to American living and working conditions. It is limited to sedentary or light muscular activity, and to rooms heated by the usual American convection methods (warm air, central fan and direct hot water and steam heating systems) in which the difference between the air and wall surface temperatures may not be great. The chart does not apply to rooms heated by radiant methods such as the British panel system, open coal fires, and the like. It will probably not apply with adequate accuracy to races other than the white or perhaps to inhabitants of other countries where the living conditions, climate, heating methods, and clothing are materially different from those of the subjects employed in experiments at the A.S.H.V.E. Research Laboratory at Pittsburgh.

The effective temperature index for persons doing medium or heavy muscular work, in still air, has also been determined at the A.S.H.V.E. Research Laboratory[10].

Example 1. Given dry-bulb and wet-bulb temperatures of 76 F and 62 F, respectively, and an air velocity of 100 fpm, determine: (1) effective temperature of the condition; (2) effective temperature with still air; (3) cooling produced by the movement of the air; (4) velocity necessary to reduce the condition to 66 deg effective temperature.

Solution. (1) In Fig. 1 draw line *AB* through given dry- and wet-bulb temperatures. Its intersection with the 100-ft velocity curve gives 69 deg for the effective temperature of the condition. (2) Follow line *AB* to the right to its intersection with the 20-fpm velocity line, and read 70.4 deg for the effective temperature for this velocity or so-called still air. (3) The cooling produced by the movement of the air is 70.4 − 69 = 1.4 deg effective temperature. (4) Follow line *AB* to the left until it crosses the 66 deg effective temperature line. Interpolate velocity value of 340 fpm, to which the movement of the air must be increased for maximum comfort.

OPTIMUM AIR CONDITIONS

No single comfort standard can be laid down which would meet every need. There is an inherent individual variation in the sensation of warmth or comfort felt by persons when exposed to an identical atmospheric condition. The state of health, age, sex, clothing, activity, and the degree of acquired adaptation seem to be the important factors affecting the comfort standards.

Since the prolonged effects of temperature, humidity and air movement on health are not known to the same extent as their effects on com-

[10]Effective Temperature for Persons Lightly Clothed and Working in Still Air, by F. C. Houghten, W. W. Teague and W. E. Miller (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926).

fort, the optimum conditions for health may not be identical with those for comfort. On general physiologic grounds, however, the two do not differ greatly since this is in accordance with the efficient operation of the heat regulating mechanism of the body. This belief is strengthened by results of studies on premature infants over a four-year period[11]. By adjusting the temperature and humidity so as to stabilize the body temperature of these infants, the incidence of diarrhœa and mortality was decreased, gains in body weight increased and infections were reduced to a minimum.

Comfort Chart; Comfort Line; Comfort Zone

Fig. 2 shows a comfort chart, developed at the A.S.H.V.E. Laboratory, on which the average and extreme comfort zones have been superimposed. The *extreme comfort zone* includes air conditions in which one or more of the experimental subjects were comfortable. The *average comfort zone* includes those air conditions in which the majority of the subjects (50 per cent or more) were comfortable. That particular effective temperature at which the maximum number of subjects was comfortable was called the *comfort line.*

The *average winter comfort zone* as determined at the A.S.H.V.E. Laboratory ranges from 63 deg to 71 deg ET (effective temperature). While at rest, 97 per cent of the experimental subjects were found to be comfortable at 66 deg ET and this temperature was accepted as the *winter comfort line* or *optimum effective temperature.*

The comfort line separates the cool air conditions to its left from the warm air conditions to its right. Under the air conditions existing along or defined by the comfort line, the body is able to maintain thermal equilibrium with its environment with the least conscious sensation to the individual, or with the minimum physiologic demand on the heat regulating mechanism. This environment involves not only the condition of the air with respect to temperature and humidity, but also the condition of the surrounding objects and wall surfaces. The comfort zone tests were made in rooms with wall surface temperatures approximately the same as the room dry-bulb temperature. For walls of large area having unusually low surface temperatures, however, a somewhat higher range of effective temperature is required to compensate for the increased loss of heat from the body by radiation[12].

The average summer comfort zone for exposures of 3 hours or more ranges from about 66 deg to 75 deg ET, based on studies made at the Harvard School of Public Health[13]. The probable optimum effective temperature (for exposures of 3 hours or more) is 71 deg. These effective temperatures average about 4 deg higher than those found in winter when customary winter clothing was worn. The variation from winter to summer is probably due partly to adaptation to seasonal weather and partly to differences in the clothing worn in the two seasons.

[11]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[12]Cold Walls and Their Relation to the Feeling of Warmth, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933, p. 53).

[13]The Summer Comfort Zone; Climate and Clothing, by C. P. Yaglou and Philip Drinker (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

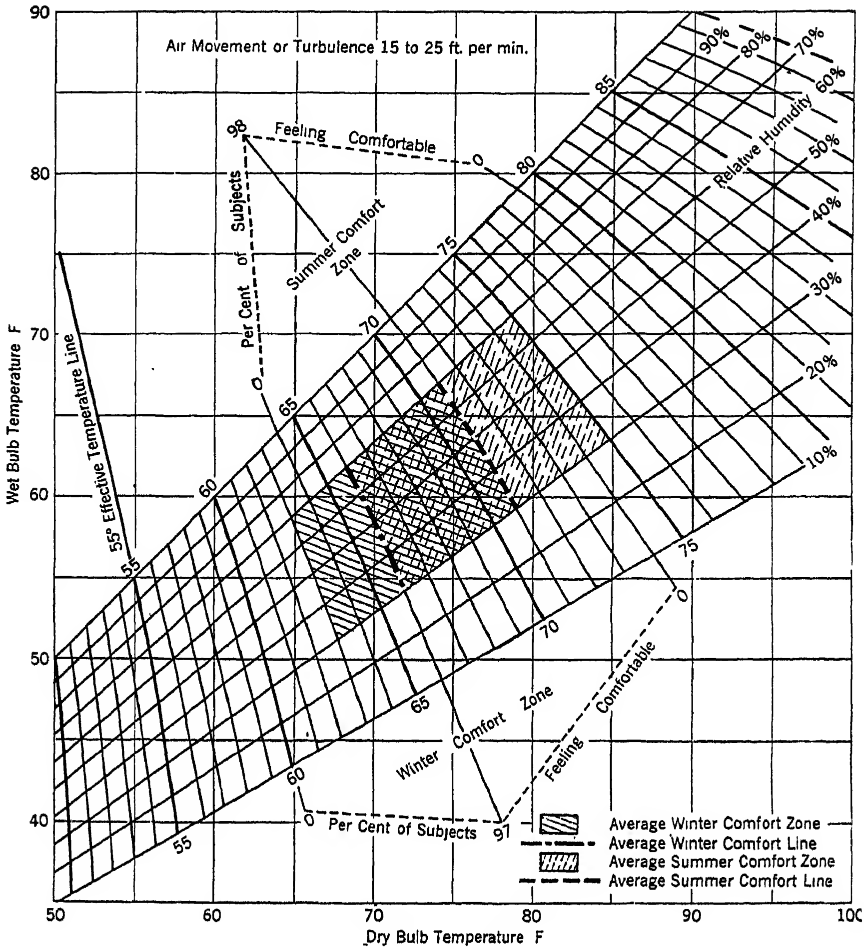

FIG. 2. A.S.H.V.E. COMFORT CHART FOR AIR VELOCITIES OF 15 TO 25 FPM (STILL AIR)[14]

Note.—Both summer and winter comfort zones apply to inhabitants of the United States only. Application of winter comfort line is further limited to rooms heated by central station systems of the convection type. The line does not apply to rooms heated by radiant methods. Application of summer comfort line is limited to homes, offices and the like, where the occupants become fully adapted to the artificial air conditions. The line does not apply to theaters, department stores, and the like where the exposure is less than 3 hours.

The best effective temperature (for exposures lasting 3 hours or more) was found to follow the average monthly outdoor temperature more closely than the prevailing outdoor temperature. It remained at approximately the same value in July, August and September, and although the average monthly temperature did not vary much, the prevailing outdoor temperature ranged from 70 F to 99.5 F. A decrease in the optimum

[14]See Report How to Use the Effective Temperature Index and Comfort Charts, (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

temperature became apparent only when the prevailing outdoor temperature fell to 66 F, which is below the customary room temperature in the United States for summer and winter.

Young men as a general rule prefer conditions in the cool region of the comfort zone, and women and older people in the warm region of the comfort zone. Crowding the experimental chamber lowered the optimum effective temperature from 70.8 deg when the gross floor area per occupant was 44 sq ft and the air space 380 cu ft, to 69.4 deg when the floor area was reduced to 14 sq ft and the air space to 120 cu ft per occupant.

Example 2. Given dry-bulb and wet-bulb temperatures of 75 and 68 F, respectively. First, what is the effective temperature? Second, is this condition warmer or cooler than 80 F dry-bulb and 60 F wet-bulb?

Solution. The first condition is given by the intersection of the 75 F dry-bulb line and the 68 F wet-bulb line (Fig. 2). The effective temperature of 72.1 deg is given by the numerical value of the effective temperature line passing through this point and indicated by the scale along the saturation curve. The second condition is given by the intersection of 80 F dry-bulb and 60 F wet-bulb and is 71.8 deg ET. It is therefore 0.3 deg ET cooler than the first condition.

Example 3. Given 76 F dry-bulb and 61 F wet-bulb, how many degrees difference between this condition and the winter comfort line or 66 deg ET?

Solution. The effective temperature for this condition is given by the intersection of the 76-F dry-bulb and 61-F wet-bulb lines and is 70 deg ET, which is 4 deg ET warmer than the comfort line.

In the comfort zone experiments of the A.S.H.V.E. Research Laboratory, the relative humidity was varied between the limits of 30 and 70 per cent approximately, but the most comfortable range has not been determined. In similar experiments at the Harvard School of Public Health, a relative humidity of 70 per cent was found to be *somewhat humid* in winter, by about half of the subjects who were stripped to the waist, even when the dry-bulb temperature was 70 F or less. In summer, a relative humidity of 30 per cent was pronounced as *a little too dry* by about a third of the subjects wearing warm-weather clothing. So long as the temperature was kept within proper limits, the majority of the subjects were unable to detect sensations of humidity (*i.e.*, too high, too low, or medium) when the relative humidity was between 30 and 60 per cent. This is in accord with studies by Howell[15], Miura[16] and others.

Dry air produces an excessive loss of moisture from the skin and respiratory tract. Owing to the cooling effect of evaporation, higher temperatures are necessary, and this condition leads to discomfort and lassitude. Moist air, on the other hand, interferes with the normal evaporation of moisture from the skin, and again may cause a feeling of oppression and lassitude, especially when the temperature is also high.

Just what the optimum range of humidity is, is a matter of conjecture. There seems to exist a general opinion, supported by some experimental and statistical data, that warm, dry air is less pleasant than air of a moderate humidity, and that it dries up the mucous membranes in such

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[15]Humidity and Comfort, by W. H. Howell (*The Science Press*, April, 1931).

[16]Effect of Variation in Relative Humidity upon Skin Temperature and Sense of Comfort, by U. Miura (*American Journal of Hygiene*, Vol. 13, 1931, p. 432).

a way as to increase susceptibility to colds and other respiratory disorders[17, 18, 19].

For the premature infant, a high relative humidity of about 65 per cent is demonstrably beneficial to health and growth[20], and according to Huntington[21], this seems to be the case for adults also. All of these studies indicate that the optimum humidity must always be considered in combination with temperature.

Until more exact information is secured, it would be desirable to restrict the comfort zones to the range of relative humidity employed in the comfort zone experiments, namely, 30 to 70 per cent. Relative humidities below 30 per cent may prove satisfactory from the standpoint of comfort, so long as extremely low humidities are avoided. From the standpoint of health, however, the consensus seems to favor a relative humidity between 40 and 60 per cent. In mild weather such comparatively high relative humidities are entirely feasible, but in cold or sub-freezing weather they are objectionable on account of condensation and frosting on the windows. They may even cause serious damage to certain building materials of the exposed walls by condensation and freezing of the moisture accumulating inside these materials. Unless special precautions are taken to properly insulate the affected surfaces, it will be necessary to reduce the degree of artificial humidification in sub-freezing weather to less than 40 per cent, according to the outdoor temperature. Information on the prevention of condensation on building surfaces is given in Chapter 7. The principles underlying humidity requirements and limitations are discussed more fully elsewhere[22].

The comfort chart (Fig. 2) applies to adults between 20 and 70 years of age living in the northeastern parts of the United States. For prematurely born infants, the optimum temperature varies from 100 F to 75 F, depending upon the stage of development. The optimum relative humidity for these infants is placed at 65 per cent. No data are yet available on the optimum air conditions for full term infants and young children up to school age. Satisfactory air conditions for these age groups are assumed to vary from 75 F to 68 F with natural indoor humidities. For school children, the studies of the *New York State Commission on Ventilation* place the optimum air conditions at 66 F to 68 F temperature with a moderate humidity (not specified) and a moderate but not excessive amount of air movement (not specified)[23].

Satisfactory comfort conditions are found to vary from 40 deg to 70 deg ET, depending upon the rate of work and amount of clothing worn. The

[17]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surface, by Mudd, Stuart, et al (*Journal Experimental Medicine*, 1921, Vol. 34, p. 11).

[18]Reactions of the Nasal Cavity and Post-Nasal Space to Chilling of the Body Surfaces, by A. Goldman, et al and Concurrent Study of Bacteriology of Nose and Throat (*Journal Infectious Diseases*, 1921, Vol. 29, p. 151).

[19]The Etiology of Acute Inflammations of the Nose, Pharynx and Tonsils, by Mudd, Stuart, et al (Am. Otol., Rinol., and Laryngol, 1921).

[20]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[21]Weather and Health, by Ellsworth Huntington (*Bulletin* of the National Research Council No. 75. The National Academy of Science, Washington, D. C., 1930).

[22]Humidification for Residences, by A. P. Kratz (*University of Illinois Engineering Experiment Station Bulletin* No. 230, July 28, 1931).

[23]Ventilation, Report of the New York State Commission on Ventilation, 1923.

effective temperatures giving maximum comfort for persons working have been determined by the A.S.H.V.E. Research Laboratory[24] for a rate of work which is considered hard labor. For this degree of work, 50 per cent were fairly comfortable for temperatures ranging from 46 to 64 deg ET, while the greatest percentage found maximum comfort at 53 deg ET. In hot industries, 80 deg ET is considered the upper limit compatible with efficiency, and, whenever possible, this should be reduced to 70 deg ET or less.

APPLICATION OF COMFORT CHART

The average winter comfort line (66 deg ET) applies to average American men and women living inside the broad geographic belt across the United States in which central heating of the convection type is generally used during four to eight months of the year. It does not apply to rooms heated by radiant energy, and has not been advocated officially for use in foreign countries where the climate, heating methods, and general living conditions are materially different from those in the United States, although several foreign workers have attempted to show that it cannot be so applied. Even in the warm south and southwestern climates, and in the very cold north-central climate of the United States, the comfort chart would probably have to be modified according to climate, living and working conditions, and the degree of acquired adaptation.

In densely occupied spaces, such as classrooms, theaters and auditoriums, somewhat lower temperatures are necessary than those indicated by the comfort line on account of counter radiation between the bodies of occupants in close proximity. In rooms in which the average wall surface temperature is considerably below the air temperature, higher air temperatures are necessary. The reverse holds true in radiant or panel heating methods. (See Chapter 37).

The sensation of comfort, insofar as the physical environment is concerned, is not absolute but varies considerably among certain individuals. Therefore, in applying the air conditions indicated by the comfort line, it should not be expected that all the occupants of a room will feel perfectly comfortable. When the winter comfort line is applied in accordance with the foregoing recommendations, the majority of the occupants will be perfectly comfortable, but there will always be a few who would feel *a bit too cool* and a few *a bit too warm*. These individual differences among the minority may be counteracted by suitable clothing.

Air conditions lying outside the average comfort zone but within the extreme comfort zone may be comfortable to certain persons. In other words, it is possible for half of the occupants of a room to be comfortable in air conditions outside the *average* comfort zone, but in the majority of cases, if not in all, these conditions will be well within the extreme comfort zone as determined experimentally.

Strictly speaking, the only authoritative comfort zone on which accurate data are available, is that for 15 to 25 fpm air movement or turbulance (often referred to as still air). In the past, the winter comfort

[24]A.S.H.V.E. research paper entitled, Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by *F. C. Houghten, W. W. Teague, W. E. Miller* and *W. P. Yant* (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

zone has often been superimposed on the thermometric chart or on effective temperature charts for various air velocities, on the assumption that air conditions of equal warmth are approximately equally comfortable. This may hold in hot industries where the workers are adapted to high temperatures and strong air currents, but it does not apply to sedentary conditions. To ascertain approximately whether a given industrial condition is reasonably comfortable, it would be necessary first to compute the effective temperature from the thermometric chart (Fig. 1) and then to refer this effective temperature to the comfort chart (Fig. 2).

The summer comfort line (71 deg ET) is applicable to the same geographic area as the winter comfort line. It is further restricted to cases in which the human body has reached thermal equilibrium with its environment. As a general rule this takes place after 1½ to 3 hours exposure. When a person from outdoors enters a room cooled to 71 deg ET on a hot day (95 F or over) an intense chill is likely to be experienced which is

TABLE 2. DESIRABLE INDOOR AIR CONDITIONS IN SUMMER CORRESPONDING TO OUTDOOR TEMPERATURES

Applicable to Exposures Less Than 3 Hours

OUTDOOR TEMP (DEG FAHR)	INDOOR AIR CONDITIONS WITH DEW-POINT CONSTANT AT 57 F		
DRY-BULB	DRY-BULB	WET-BULB	EFFECTIVE TEMP
95	80.0	65.0	73
90	78.0	64.5	72
85	76.5	64.0	71
80	75.0	63.5	70
75	73.5	63.0	69
70	72.0	62.5	68

unpleasant. However, after remaining in the room for about 2 hours, this fundamental optimum condition will prove satisfactory to the average person. The summer comfort zone, as well as the comfort line, makes proper allowance for these adaptive changes in the body, and thus applies to homes, offices, schools and other similar places where persons of sedentary occupations spend from 3 to 8 or more hours daily.

In artificially cooled theaters, department stores, restaurants, and other public buildings where the period of occupancy is short, the contrast between outdoor and indoor air conditions becomes the deciding factor in regard to the temperature and humidity to be maintained. The object of cooling such places in the summer is not to reduce the temperature to the optimum degree, but to maintain therein a temperature which is temporarily comfortable to the patrons who thus avoid sensations of chill and intense heat on entering and leaving the building. The relative humidity should be low enough (about 50 per cent or less) to give a sense of comfort without chill and to induce a rate of evaporation which will keep clothing and skin dry. For exposures less than 3 hours, desirable indoor conditions in summer corresponding to various outdoor temperatures are given in Table 2.

A.S.H.V.E. VENTILATION STANDARDS[25]

It is the intent of the Committee in presenting this report to confine itself to a statement of those requirements which, based on present day knowledge, will provide adequate ventilation for spaces intended for human occupancy. The following standards shall apply to all spaces occupied by human beings in all buildings for which ventilation regulations are to be established.

SECTION I—AIR TEMPERATURE AND HUMIDITY

The *temperature* and *humidity* of the air in such occupied spaces, and in which the only source of contamination is the occupant, shall be maintained at all times during occupancy at an *Effective Temperature*, as hereinafter stated.

The relative humidity shall be not less than 30 per cent, nor more than 60 per cent in any case. The Effective Temperature shall range between 64 deg and 69 deg when heating or humidification is required, and between 69 deg and 73 deg when cooling or dehumidification is required.

These Effective Temperatures shall be maintained at a level of 36 in. above the floor. (See Appendix, Tables A and B).

SECTION II—AIR QUALITY

The air in such occupied spaces shall at all times be free from toxic, unhealthful or disagreeable gases and fumes and shall be relatively free from odors and dust.

In every space coming within the provisions of these requirements and in which the quality of the air is below the standards prescribed by good medical and engineering practices, due to toxic substances, bacteria, dust, excessive temperature, excessive humidity, objectionable odors, or other similar causes, means for ventilating shall be provided so that the quality of the air shall be raised to these standards.

SECTION III—AIR MOTION

The air in such occupied spaces shall at all times be in constant motion sufficient to maintain a reasonable uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces.

The air motion in such occupied spaces, and in which the only source of contamination is the occupant, shall have a velocity of not more than 50 feet per minute, measured at a height of 36 in. above the floor.

SECTION IV—AIR DISTRIBUTION

The air in all rooms and enclosed spaces shall, under the provisions of these requirements, be distributed with reasonable uniformity, and the variation in the carbon dioxide content of the air shall be taken as a measure of such distribution.

The air in a space ventilated in accordance with these requirements, and in which the only source of contamination is the occupant, shall be distributed and circulated so that the variation in the concentration of carbon dioxide, when measured at a height of 36 in. above the floor, shall not exceed one part in 10,000.

SECTION V—AIR QUANTITY

The quantity of air used to ventilate the given space during occupancy shall always be sufficient to maintain the standards of air temperature, air quality, air motion and air distribution as herein required. Not less than 10 cubic feet per minute per occupant of the total air circulated to meet these requirements shall be taken from an outdoor source.

APPENDIX

Definitions

For the purposes of these standards the terms used shall be defined as follows:—

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*).

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[25]Report of A.S.H.V.E. Committee on Ventilation Standards consisting of W. H. Driscoll, *Chairman*, J. J. Aeberly, F. Paul Anderson, L. A. Harding, D. D. Kimball, J. R. McColl, C. L. Riley, W. A. Rowe, Perry West and A. C. Willard, presented at the Semi-Annual Meeting of the Society, Milwaukee, Wis., June, 1932, and adopted by the Society in August, 1932.

TABLE A. EFFECTIVE TEMPERATURES RANGING FROM 64 DEG TO 69 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a]

(For use when heating or humidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
67						64.0	64.3
68			64.0	64.2	64.5	64.8	65.1
69	64.1	64.4	64.8	65.1	65.4	65.7	66.0
70	64.8	65.1	65.4	65.8	66.2	66.5	66.8
71	65.5	65.8	66.2	66.6	67.0	67.3	67.7
72	66.2	66.5	66.9	67.3	67.7	68.1	68.5
73	67.0	67.3	67.7	68.1	68.5	68.9	
74	67.7	68.0	68.4	68.8			
75	68.4	68.7					
76	69.0						

[a]See Fig. 2, p. 29.

TABLE B. EFFECTIVE TEMPERATURES RANGING FROM 69 DEG TO 73 DEG FOR VARIOUS DRY-BULB TEMPERATURES AND RELATIVE HUMIDITIES FOR STILL AIR FOR PERSONS NORMALLY CLOTHED AND SLIGHTLY ACTIVE[a-b]

(For use when cooling or dehumidification is required)

DRY-BULB TEMPERATURES (DEG FAHR)	RELATIVE HUMIDITIES (PER CENT)						
	30	35	40	45	50	55	60
	EFFECTIVE TEMPERATURES (DEGREES)						
73							69.3
74					69.3	69.7	70.1
75			69.1	69.5	70.0	71.5	71.0
76	69.0	69.4	69.9	70.5	70.8	71.3	71.8
77	69.7	70.2	70.7	71.2	71.6	72.1	72.6
78	70.4	70.9	71.4	71.9	72.4	73.0	
79	71.1	71.6	72.2	72.6			
80	71.8	72.4	72.9				
81	72.5						

[a]See Fig. 2, p. 29.

[b]This table applies primarily to cases in which the human body has reached equilibrium with the surrounding air. A higher plane of summer effective temperatures is required in places of public assembly where the period of occupancy is short, than is required for offices and industrial plants where the period of occupancy is of longer duration. When the period of occupancy is two hours or less, the dry-bulb temperature shall be 72 F plus one-third of the difference between the outside dry-bulb temperature and 70 F, and the relative humidity shall not exceed 60 per cent. (See also Table 2, p. 29).

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Dry-Bulb Temperature: The temperature of the air which is indicated by any type of thermometer which is not affected by the water vapor content or relative humidity of the air.

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, such as are blown on a windy day, the average diameter of which is approximately 0.01 centimeter, may be called dust.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Humidity: The water vapor (either saturated or superheated steam) occupying any space, which may or may not contain other vapors and gases at the same time.

Relative Humidity: A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor.

Spaces in Which the Only Source of Contamination Is the Occupant: Spaces in which the atmospheric contamination results entirely from the respiratory processes of the occupant, including heat, moisture, and odors given off by the body. No manufacturing or industrial processes or other sources of atmospheric contamination, including heat and moisture, than people are considered under this title.

FACTORS INFLUENCING APPLICATIONS

The conditions and limitations outlined under the heading Application of Comfort Chart should be noted in applying the temperatures and relative humidities specified in Tables *A* and *B* of the preceding A.S.H.V.E. Ventilation Standards.

Air Quality

In occupied spaces in which the vitiation is entirely of human origin, the chemical composition of the air, the dust, and bacteria content may be dismissed from consideration so that the problem consists in maintaining a suitable temperature with a moderate humidity, and in keeping the atmosphere free from objectionable odors. Such unpleasant odors, human or otherwise, can be easily detected by persons entering the room from clean, odorless air. A further discussion of air quality will be found in Chapters 15 and 16.

Air Motion

The air in occupied spaces must be in constant gentle motion sufficient to maintain a satisfactory uniformity of temperature and humidity, but not such as to cause objectionable drafts in any occupied portion of such spaces. Stagnant air, no matter how pure, is depressing, and it fails to produce the pleasant and stimulating effect of cool air in gentle motion.

Studies by Baetjer[26] on the influence of air motion on comfort indicate that in ordinary air conditioning work the velocity of air currents should never be allowed to fall below 5 fpm, nor should it be allowed to exceed 50 fpm, except when the temperature of the air current striking the face is higher than the temperature of the room. The lower limit of 5 fpm may be taken as the minimum during the heating season, and the upper limit of 50 fpm as the maximum during the cooling season.

Air Distribution[27]

As a rule satisfactory distribution is secured when the air movement or turbulence as measured by the Kata thermometer (see p. 573, Chapter 40) is uniform in all parts of the occupied space and when simultaneous readings of temperature at any two points on the same level within the occupied space do not differ by more than 3 deg. Measurements of CO_2 are acceptable in lieu of temperature and air motion variations, but the usual method of determining CO_2 in air is much more laborious than the

[26]Threshold Air Currents in Ventilation (*American Journal of Hygiene*, Vol. IV, No. 6, p. 650, 1924).

[27]Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933, p. 324).

determination of temperature and air movement. When CO_2 is used as an index of distribution, the variation in the concentration of the gas at a height of 36 in. above the floor should not exceed one part in 10,000 parts of air.

Air Quantity

The quantity of air to be circulated through an occupied space, whether by natural or mechanical means, or whether the air is conditioned or not, must in all cases be sufficient to maintain the required standards of air temperature, quality, motion and distribution. The factors which determine air quantity include the type and nature of the building, locality, climate, height of rooms, floor area, window area, extent of occupancy, and last but not least, the method of distribution.

Actually there are two air quantities to be considered, namely, (1) total air required and (2) outside air required. The difference between these two quantities represents the amount to be recirculated, or

$$\text{Total air} = \text{outside air} + \text{recirculated air.}$$

Sometimes the ratio of the outside air to the total air can be decreased if the air introduced is conditioned, but in the light of present information, a minimum of 10 cfm of outdoor air per person should be provided. If the air is not conditioned as in a ventilating system, the vitiated air is usually exhausted to the atmosphere, in which case all of the air introduced is outside air.

Temperature Rise

The total quantity of air introduced is governed largely by the allowable *temperature rise* when cooling is required and the allowable *temperature drop* when heating is required. As a rule, the introduction and distribution of warm air into an occupied space does not present as many difficulties as does the introduction of cold air. The former is determined from the amount of heat to be given up to the space, and the latter is determined from the amount of heat to be removed from the space, using a temperature rise that will produce uniform distribution without the production of disagreeable drafts.

Two of the most important factors on which the *temperature rise* depends are (1) the method of distribution and (2) the most economical temperature rise for the conditions involved. Some systems of distribution produce drafts with but a few degrees temperature rise, while other systems operate successfully with a temperature rise as high as 35 deg. The total air quantity introduced in any particular case is inversely proportional to the temperature rise, and depends largely upon the judgment and ingenuity of the engineer in designing the most suitable system for the particular conditions. Small quantities of air reduce the size of equipment, ducts, space, and initial cost, but require lower air temperatures. In any specific case, the cost of refrigeration must be balanced against the extra cost in increased size of equipment and running expense.

Outside Air. In order to provide uniform temperature conditions, it

is necessary to maintain a pressure of about 0.1 in. of water in the room or space to be ventilated or conditioned. This usually requires the introduction of a certain amount of outside air which depends on the particular conditions involved, and may vary over a considerable range.

In rooms in which the only source of contamination is the occupant the minimum quantity of outside or *new* air to be circulated appears to be that necessary to remove objectionable *body odors*. The concentration of body odors in turn depends largely upon the temperature of the air; the higher the temperature, the greater the amount of perspiration (sensible or insensible) given off from the skin, and the greater the concentration of odors.

Under proper temperature conditions, body odors may be reduced to a concentration that is not objectionable by as little as 10 cfm of outside air per person. This is the minimum amount specified in the Ventilation Standards adopted in 1932 by the Society. The ventilation laws of many states require the introduction of 30 cfm of outside air per occupant, but the present tendency is to supply a smaller amount of conditioned air.

The total quantity of air required to maintain the standards of temperature, distribution, and air motion is usually at least twice as great as that required to keep down body odors, owing largely to difficulties encountered with distribution systems.

NATURAL AND MECHANICAL VENTILATION

Under favorable conditions natural ventilation methods properly combined with means for heating may be sufficient to provide for the foregoing standards. As a rule, in instances in which the only source of contamination is the occupant, the requirements may be fulfilled when the following conditions prevail:

1. At least 50 sq ft of floor area for each occupant.
2. At least 500 cu ft of air space per occupant.
3. Effective openings in windows and skylights equal to at least 5 per cent of the floor area.

Whenever natural means are not sufficient to maintain the standards, resort must be made to whatever modifications or mechanical apparatus are necessary to secure such standards.

In large offices, large school rooms, and in public and industrial buildings, natural ventilation is uncertain and makes heating difficult. The chief disadvantage of natural methods is the lack of control: they depend largely on weather and upon the velocity and direction of the wind. Rooms on the windward side of a building may be difficult to heat and ventilate on account of drafts, while rooms on the leeward side may not receive an adequate amount of air from out of doors. The partial vacuum produced on the leeward side under the action of the wind may even reverse the flow of air so that the leeward half of the building has to take the *drift* of the air from the rooms of the windward half. Under such conditions no outdoor air would enter through a leeward window opening, but room air would pass out.

In warm weather natural methods of ventilation afford little or no

control of indoor temperature and humidity. Outdoor smoke, dust and noise, constitute other limitations of natural methods.

RECIRCULATION

The saving in operating costs due to recirculation of the air, while very considerable, must not be obtained at the expense of air quality. The percentage of recirculated air may be varied to suit the seasonal changes so as to conserve heat in winter and refrigeration in summer, but at no time during occupancy should there be taken from out of doors less than 10 cfm for each occupant. As a general rule, recirculation impairs the quality of the air by excessive humidity (if not conditioned), excessive odors, or both, and it tends to deprive the air of its ionic content[28], but the influence of this factor on comfort and health is at present a matter of speculation.

Toilets and similar rooms and all kitchens in buildings using recirculation should be separately, mechanically ventilated, with the exhaust in excess of the supply, in order to prevent objectionable odors from diffusing into other parts of the building. This air removal may in many cases be sufficient to insure an adequate replacement of outside air to the general recirculating system.

OZONE

The value of ozone in recirculated air has been greatly exaggerated in the past. Numerous researches[29], [30], [31] have shown that ozone in concentrations permissible in air conditioning work (0.1 to 0.5 parts per million parts of air) exerts practically no effect on pathogenic air-borne organisms and that it does not destroy the source of odors. Ozone, however, may mask odors by olfactory compensation. It requires at least 13 parts of ozone per million parts of air to influence bacteria[32]. Human beings are injuriously affected by ozone in concentrations of one part or more per million parts of air[32]. This amount will not destroy odors nor kill bacteria.

ULTRA-VIOLET RADIATION AND IONIZATION

In spite of rapid advances in air conditioning during the past few years, the secrets of reproducing indoors the natural climatic elements as they exist in open country under ideal weather conditions have not as yet been fully ascertained. Extensive studies have failed to discover the stimulating quality in open country air which is lost when the air is brought indoors, and particularly when it is treated mechanically. Ultra-violet light and ionization have been suggested but have not yet been identified.

It is generally recognized that total solar radiation in open air is the

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[28]See A.S.H.V.E. research paper entitled, Changes in Ionic Content in Occupied Rooms Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. Transactions, Vol. 37, 1931).

[29]Proceedings, Royal Society of London, by L. Hill and M. Flack (1911, B, Vol. 84, p. 404).

[30]Jordan and Carlson (*Journal American Medical Association*, 1913, Vol. 51, p. 1007).

[31]Konrich (Ztschr. f. Hyg., 1913, Vol. 73, p. 443).

[32]Preventive Medicine and Hygiene, by Milton J. Rosenau.

greatest curative and is a powerful germicidal agent. A critical review of the literature, however, discloses that artificial ultra-violet radiation has little importance in air conditioning, at least for the time being, because it is not known which of the solar rays brings about the cure, and what physiologic processes are involved in their action. With the exception of rickets and certain skin diseases, artificial radiation was found to have no effect on susceptibility, incidence, and resistance to respiratory infections[33], no permanent effects on blood and no effect on the general metabolism of human beings.

Ionization, on the other hand, seems to offer a fruitful field for research.

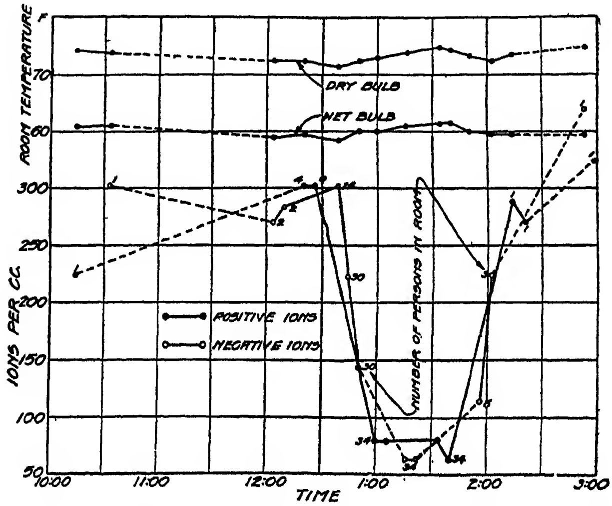

FIG. 3. INFLUENCE OF ROOM OCCUPANCY ON IONIC CONTENT[34]

(Cubical Contents of Room, 10,000 Cu Ft; Number of Occupants, 34)

Recent experiments[34] show that in occupied rooms there is a marked decrease in both positive and negative small ions. As shown in Fig. 3, soon after the occupants assembled the ionic content fell abruptly to a very low level which was maintained until the occupants left the room. Both positive and negative ions began to rise again as soon as the occupants departed. The problem now is to determine whether such alterations in the electrical quality of air have any significant bearing on comfort and health. If this proves to be the case, some artificial source of ionization may be desirable in occupied rooms.

[33]Light—Its Photodynamic Activity and Use as a Therapeutic Agent, by F. W. Schultz, K. W. Stenstrom and E. M. Clausen (Report of the White House Conference on Child Health and Protection, Washington, February, 1931).

[34]A.S.H.V.E. research paper entitled, Changes in Ionic Content in Occupied Rooms, Ventilated by Natural and Mechanical Methods, by C. P. Yaglou, L. C. Benjamin and S. P. Choate (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931). Physiologic Changes During Exposure to Ionized Air, by C. P. Yaglou, A. D. Brandt and L. C. Benjamin (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, August, 1933). Diurnal and Seasonal Variations in the Small Ion Content of Outdoor and Indoor Air, by C. P. Yaglou and L. C. Benjamin, (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1934).

HEAT AND MOISTURE LOSSES

In order to solve air conditioning problems involving the human body it is necessary to know the rate at which sensible and latent heat are given up by the body under various conditions of temperature and activity. Research at the A.S.H.V.E. Laboratory[35] has resulted in the data given in Figs. 4, 5, 6 and 7. Table 3 gives the metabolic rates for various degrees of activity.

The experimental data from which the curves were drawn indicate that

TABLE 3. RELATION BETWEEN METABOLIC RATE AND ACTIVITY[35]

ACTIVITY	METABOLIC RATE BTU PER HOUR FOR AVERAGE MAN (19.5 SQ FT SURFACE AREA)	AUTHORITY
Seated at rest	384	Research Laboratory, American Society of Heating and Ventilating Engineers.
Standing at rest	431	Research Laboratory, American Society of Heating and Ventilating Engineers.
Walking 2 mph	761	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 3 mph	1049	Douglas, Haldane, Henderson and Schneider
Walking 4 mph	1388	Average values from Douglas, Haldane, Henderson and Schneider; and Henderson and Haggard.
Walking 5 mph	2530	Douglas, Haldane, Henderson and Schneider
Slow run	2285	Henderson and Haggard
Very severe exercise	2555	Benedict and Carpenter
Maximum exertion	3333 to 4762+	Henderson and Haggard
Tailor	482	Becker and Hamalainen
Bookbinder	626	Becker and Hamalainen
Shoemaker	661	Becker and Hamalainen
Carpenter	762 to 963	Becker and Hamalainen
Metal Worker	862	Becker and Hamalainen
Painter (of furniture)	876	Becker and Hamalainen
Stonemason	1488	Becker and Hamalainen
Man sawing wood	1797	Becker and Hamalainen

total heat loss does not vary appreciably within the comfort zone range (see Fig. 4). Above or below this range the variation seems to be approximately a function of effective temperature. Sensible and latent heat losses (Figs. 5 and 7) on the other hand, vary greatly within the comfort zone range, the variation following more closely the dry-bulb temperature than any other factor.

Although total heat loss and sensible and latent heat losses are not exact functions of effective and dry-bulb temperature, respectively, for all conditions of humidity and air motion, they are plotted as such in the curves. This is accomplished by approximations which are sufficiently accurate for application to practical problems.

An atmospheric condition resulting in sensible perspiration is to be

[35]See A.S.H.V.E. research paper entitled Heat and Moisture Losses from Men at Work and Application to Air Conditioning Problems, by F. C. Houghten, W. W. Teague, W. E. Miller and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

avoided for obvious reasons. Tables 4 and 5 give the approximate effective temperatures at which perspiration is noticeable in different degrees for 95 per cent and 20 per cent relative humidity.

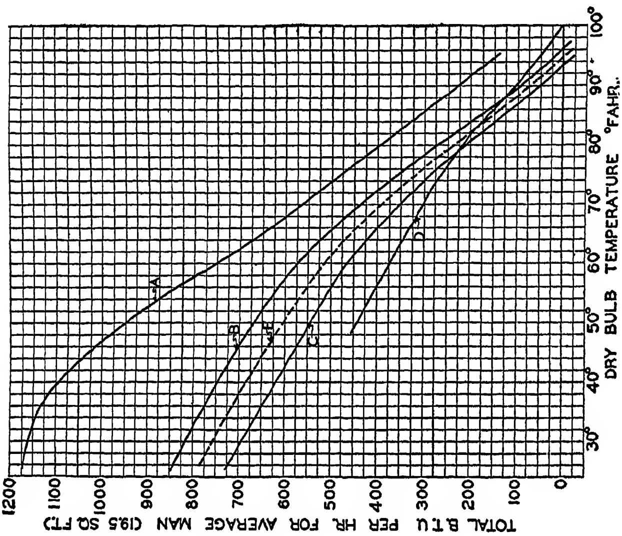

FIG. 5. RELATION BETWEEN SENSIBLE HEAT LOSS FROM THE HUMAN BODY AND DRY-BULB TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

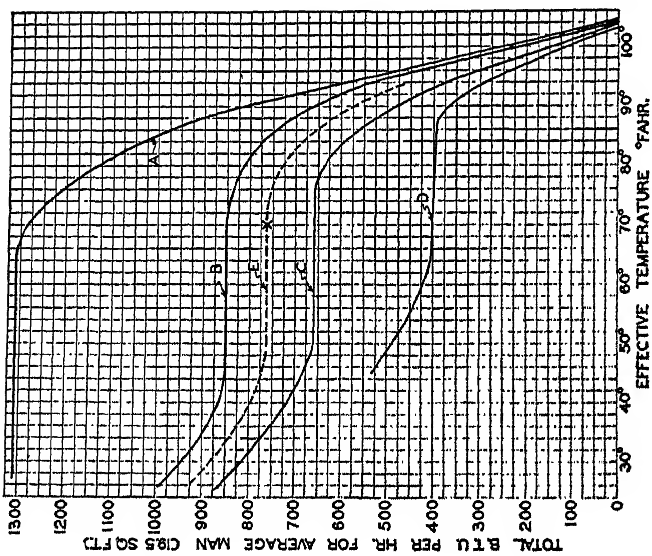

FIG. 4. RELATION BETWEEN TOTAL HEAT LOSS FROM THE HUMAN BODY AND EFFECTIVE TEMPERATURE FOR STILL AIR[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curve *A* and *C* drawn from data at an effective temperature of 70 deg only and extrapolating the relation between curves *B* and *D*, which were drawn from data at many temperatures.

In theaters, auditoriums, department stores and other crowded enclosures, the amount of heat and moisture given off by the people is so large that normal changes in outside temperature and humidity have relatively little effect on indoor air conditions. The principal object of air

conditioning in such places is to remove excessive heat and moisture by supplying a sufficient quantity of properly conditioned air. The indoor air conditions, however, must be varied according to the outside temperature, as already pointed out.

Although heat and moisture from the human body constitute the major

TABLE 4. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS SEATED AT REST UNDER VARIOUS ATMOSPHERIC CONDITIONS[36]

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B	W. B.
Forehead clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body clammy	**73.0**	73.6	72.4	**75.0**	87.0	60.7
Body damp	**79.0**	79.7	78.4	**81.0**	97.5	67.5
Beads on forehead	**80.0**	80.8	79.4	**87.0**	109.4	75.2
Body wet	**84.5**	85.4	84.0	**86.5**	108.5	74.6
Perspiration on forehead runs and drips	**88.0**	89.0	87.6	**94.0**	125.2	85.4
Perspiration runs down body	**88.5**	89.5	88.1	**90.0**	116.0	79.5

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

portion of the cooling load, in most cases where air conditioning is provided for comfort and health other factors must also be considered. These include heat from lights, machinery, and processes, as well as the transmission and infiltration of heat through the building structure. The computations for these factors may be made in accordance with data given in Chapters 5 and 7.

In many cases, allowance must also be made for sun effect and for heat capacity of the building structure in accordance with studies by the

TABLE 5. CONDITION OF SENSIBLE PERSPIRATION FOR PERSONS AT WORK UNDER VARIOUS ATMOSPHERIC CONDITIONS

DEGREE OF PERSPIRATION[a]	ATMOSPHERIC CONDITION					
	95 Per Cent Relative Humidity			20 Per Cent Relative Humidity		
	E. T.	D. B.	W. B.	E. T.	D. B.	W. B.
Forehead clammy	**59.0**	59.4	58.3	**69.5**	80.5	56.5
Body clammy	**50.0**	50.2	49.3	**57.0**	61.6	44.2
Body damp	**60.0**	60.3	59.3	**62.5**	69.6	49.5
Beads on forehead	**68.0**	68.5	67.5	**76.0**	91.0	63.4
Body wet	**69.0**	69.6	68.5	**71.0**	82.8	53.0
Perspiration on forehead runs and drips	**78.5**	79.3	78.0	**82.0**	100.5	70.2
Perspiration runs down body	**79.0**	79.8	78.5	**81.0**	99.8	69.0

[a]Forty per cent of subjects registered degree of perspiration equal to or greater than indicated.

[36]Thermal Exchanges between the Human Body and its Atmospheric Environment, by F. C. Houghten, W. W. Teague, W. E. Miller, W. P. Yant (*American Journal Physiology*, Vol. 88, No. 3, April, 1929, pp. 386-406).

A.S.H.V.E. Research Laboratory[37]. Another item to be considered is the radiant heat received by the body from high temperature wall and ceiling surfaces.

Example 4. Assume that the design of an air conditioning system for a theater is to be based on an outdoor dry-bulb temperature of 95 F and a wet-bulb temperature of 78 F with an indoor relative humidity of 50 per cent. According to Table 2, the dry-bulb temperature in the auditorium should be 80 F. Estimate the sensible and latent heat given up per person.

Solution. The sensible heat given up per person per hour under this condition may be obtained from Fig. 5. With an abscissa value of 80 F, Curve *D* for men seated at rest

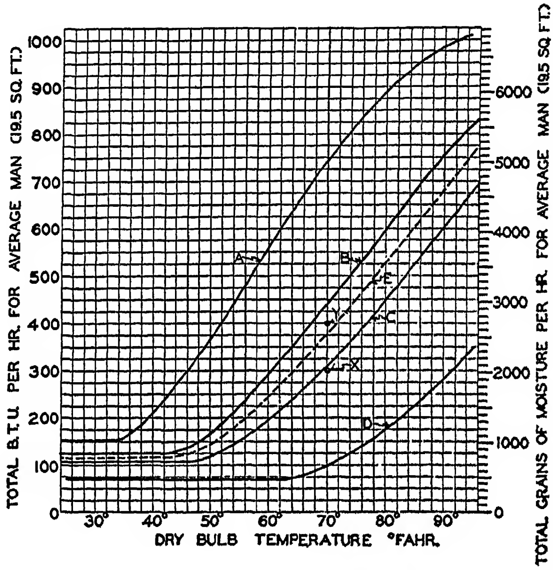

FIG. 6. LATENT HEAT AND MOISTURE LOSS FROM THE HUMAN BODY BY EVAPORATION, IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

gives a value on the ordinate scale of 220 Btu per person per hour as the sensible heat loss. The latent heat given up by a person seated at rest per hour may be obtained from Fig. 6. With an abscissa value of 80 F, Curve *D* indicates a latent heat loss of 175 Btu per hour (left hand scale) or a moisture loss of 1190 grains per hour (right hand scale).

Example 5. How much sensible heat, how much latent heat and how much water vapor will be added per hour to the atmosphere of an auditorium by an audience of 1000 adults, when the dry- and wet-bulb temperatures are 75 F and 63.5 F, respectively?

Solution. From Curve *D*, Fig. 5, find the sensible heat loss per person for a dry-bulb temperature of 75 F and still air to be 265 Btu per hour. From Fig. 6 find the latent heat loss per person for a dry-bulb temperature of 75 F to be 134 Btu per hour and the moisture added to be 905 grains per hour. Sensible heat = 1,000 × 265 = 265,000 Btu. Latent heat = 1,000 × 134 = 134,000 Btu. Water vapor added per hour to the air in the auditorium = 1,000 × 905 = 905,000 grains or 129 lb.

[37]See A.S.H.V.E. research paper entitled, Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

The sensible and latent heat added to the air may also be found as follows: The effective temperature for dry- and wet-bulb temperatures of 75 F and 63.5 F, respectively, is 70.3 deg. From Curve *D*, Fig. 4, find 403 Btu as the total heat added to the air by a person for an effective temperature of 70.3 deg. From Fig. 7 find the percentage of sensible and latent heat at a dry-bulb temperature of 75 F to be 66.5 per cent and 33.5 per cent. The sensible heat added to the air in the auditorium is 1,000 × 0.665 × 403 = 267,995 Btu per hour. The latent heat added is 1,000 × 0.335 × 403 = 135,005 Btu per hour.

Example 6. If the dry- and wet-bulb temperatures of the auditorium were 85 F and 63 F, respectively, how much heat and moisture would be dissipated to the atmosphere?

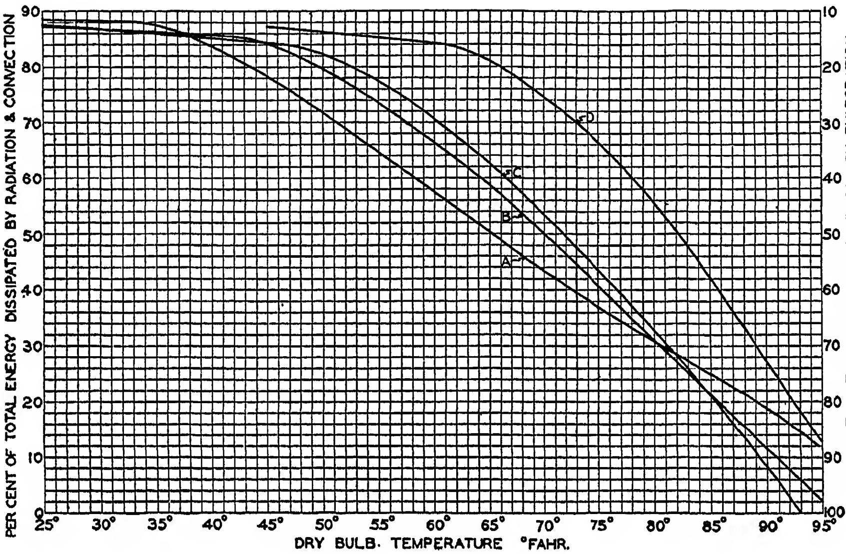

FIG. 7. HEAT LOSS FROM THE HUMAN BODY BY EVAPORATION, RADIATION AND CONVECTION IN RELATION TO DRY-BULB TEMPERATURE FOR STILL AIR CONDITIONS[a]

[a]Curve *A*—Men working 66,150 ft-lb per hour. Curve *B*—Men working 33,075 ft-lb per hour. Curve *C*—Men working 16,538 ft-lb per hour. Curve *D*—Men seated at rest. Curves *A* and *C* drawn from data at a dry-bulb temperature of 81.3 F only and extrapolating the relation between curves *B* and *D* which were drawn from data at many temperatures.

Solution. From Figs. 5 and 6, respectively, the sensible and latent heat losses per person for a dry-bulb temperature of 85 F are found to be 164 and 225 Btu per hour. The water vapor added to the atmosphere is 1,520 grains per hour. The audience will then add 164,000 Btu sensible heat, 225,000 Btu latent heat and 1,520,000 grains or 217 lb of water vapor to the air in the auditorium per hour.

Examples 5 and 6 demonstrate that while the effective temperature, and hence the feeling of warmth and rate of total heat loss, do not differ greatly in the two cases, the relative proportion of sensible and latent heat loss is reversed. In order to maintain the air conditions stipulated the air conditioning equipment must remove 63.4 per cent more sensible heat in Example 5 than in Example 6; and 67.9 per cent more latent heat or water vapor in Example 6 than in Example 5. In Example 5, 66.3 per cent of the total heat loss is sensible while in Example 6 only 42.2 per cent of the total loss is sensible.

Example 7. Neglecting the gain or loss of heat to an auditorium by transmission or infiltration through the walls, windows and doors, how many cubic feet of outside air,

with dry- and wet-bulb temperatures of 65 F and 59 F, respectively, (63.1 deg ET) must be supplied per hour to an auditorium containing 1000 people in order that the inside shall not exceed 75 F (dry-bulb) and 65 F (wet-bulb), respectively?

Solution. Figs. 5 and 6 give 265 Btu sensible heat and 905 grains of moisture as the additions per person with a dry-bulb temperature of 75 F in the auditorium. Therefore 265,000 Btu of sensible heat and 905,000 grains of moisture will be added to the air in the auditorium per hour.

Taking 0.24 as the specific heat of air, 2.4 Btu per pound of air will be required to raise the dry-bulb temperature from 65 to 75 F and $\frac{265,000}{2.4} = 110,400$ lb of air or $110,400 \times 13.4 = 1,479,000$ cfh of air will be required. This is equivalent to $\frac{1,479,000}{1000 \times 60} = 24.7$ cfm per person.

The moisture content of the inside air as taken from a psychrometric chart is 76 grains per pound of dry air and that of the outside condition is 65 grains. The increase in moisture content will therefore be 11 grains per pound of dry air. Hence $\frac{905,000}{11.0} = 82,300$ lb of air at the specified condition will be required. This is equivalent to $82,300 \times 13.4 = 1,103,000$ cfh of air or $\frac{1,103,000}{1000 \times 60} = 18.4$ cfm of air per person.

The higher volume of 24.7 cfm per person will be required to keep the dry-bulb temperature from rising above the 75 F specified. The wet-bulb temperature will therefore not rise to the maximum of 65 F.

Example 8. Assume that a man performs work at a rate equivalent to 50,000 ft-lb per hour, in an atmosphere having a dry-bulb temperature of 70 F. Estimate the sensible and latent heat given off per hour.

Solution. Since the net mechanical efficiency of the human body is about 20 per cent, the increase in metabolism due to work, over the resting metabolism, will be $\frac{50,000}{778^* \times 0.20} = 320$ Btu per hour. Assuming a resting metabolism of 400 Btu per hour (see Fig. 4), the total metabolism during work will be $400 + 320 = 720$ Btu per hour, and the total heat loss $720 - \frac{50,000}{778} = 656$ Btu per hour approximately. In Fig. 7, follow a vertical line from a dry-bulb temperature of 70 F to a point midway between Curves *A* and *B*. The sensible heat loss is about 46 per cent of the total loss, or $0.46 \times 656 = 302$ Btu per hour, and the latent heat 54 per cent of the total or $0.54 \times 656 = 354$ Btu per hour.

In cases in which the external work is not appreciable, as for instance in sewing, walking on a level road, and the like, the rate of heat loss will be approximately equal to the total metabolism.

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*Heat equivalent of mechanical work in foot-pounds per Btu.

Chapter 3

INDUSTRIAL AIR CONDITIONING

Moisture Content and Regain, Hygroscopic Materials, Atmospheric Conditions Required, Air Conditioning of Libraries, Banana Ripening, Greenhouse Heating, Apparatus for Industrial Conditioning, Industrial Humidifying Systems, Direct Humidifiers, Combined Direct and Indirect Humidifiers, Dehumidifiers

IN many industries, the temperature and relative humidity of the air have a marked influence upon the rate of production and the weight, strength, appearance, and general quality of the product. These results are due to the fact that most materials of animal or vegetable origin, and to a lesser extent minerals in certain forms, either take up or give moisture to the surrounding air. Air conditioning is applicable to industrial or process conditioning for the improvement of products during manufacture, or for making the process independent of climatic conditions.

MOISTURE CONTENT AND REGAIN

The terms *moisture content and regain* refer to the amount of moisture in hygroscopic materials. *Moisture content* is the more general term and refers either to free moisture (as in a sponge) or to hygroscopic moisture (which varies with atmospheric conditions). It is usually expressed as a percentage of the total weight of material. *Regain* is more specific and refers only to hygroscopic moisture. It is expressed as a percentage of the *bone-dry* weight of material. For example, if a sample of cloth weighing 100.0 grains, is dried to a constant weight of 93.0 grains, the loss in weight, or 7.0 grains, represents the weight of moisture originally contained. This expressed as a percentage of the total weight (100.0 grains) gives the moisture content or 7 per cent. The regain, which is expressed as a percentage of the bone-dry weight, is $\frac{7.0}{93.0}$ or 7.5 per cent.

The use of the term *regain* does not necessarily imply that the material as a whole has been completely dried out and has re-absorbed moisture. In the case of certain textiles, for instance, complete drying during manufacturing is avoided as it might appreciably reduce the ability of the material to re-absorb moisture. In measuring moisture it is necessary to dry out a sample so that the loss in weight may be used as a basis for calculating the regain of the whole lot.

HYGROSCOPIC MATERIALS

Air conditioning is extensively used in the manufacture or processing of hygroscopic materials such as textiles, paper, wood, leather, tobacco and

TABLE 1. REGAIN OF HYGROSCOPIC MATERIALS

Moisture Content Expressed in Per Cent of Dry Weight of the Substance at Various Relative Humidities—Temperature, 75 F

CLASSIFICATION	MATERIAL	DESCRIPTION	RELATIVE HUMIDITY—PER CENT									AUTHORITY
			10	20	30	40	50	60	70	80	90	
Natural Textile Fibres	Cotton	Sea Island—Roving	2.5	3.7	4.6	5.5	6.6	7.9	9.5	11.5	14.1	Hartshorne
	Cotton	American—Cloth	2.6	3.7	4.4	5.2	5.9	6.8	8.1	10.0	14.3	Schloesing
	Cotton	Absorbent	4.8	9.0	12.5	15.7	18.5	20.8	22.8	24.3	25.8	Fuwa
	Wool	Australian Merino—Skein	4.7	7.0	8.9	10.8	12.8	14.9	17.2	19.9	23.4	Hartshorne
	Silk	Raw Chevennes—Skein	3.2	5.5	6.9	8.0	8.9	10.2	11.9	14.3	18.8	Schloesing
	Linen	Table Cloth	1.9	2.9	3.6	4.3	5.1	6.1	7.0	8.4	10.2	Atkinson
	Linen	Dry Spun—Yarn	3.6	5.4	6.5	7.3	8.1	8.9	9.8	11.2	13.8	Sommer
	Jute	Average of Several Grades	3.1	5.2	6.9	8.5	10.2	12.2	14.4	17.1	20.2	Storch
	Hemp	Manila and Sisal—Rope	2.7	4.7	6.0	7.2	8.5	9.9	11.6	13.6	15.7	Fuwa
Rayons	Viscose Nitrocellulose Cupramonium	Average Skein	4.0	5.7	6.8	7.9	9.2	10.8	12.4	14.2	16.0	Robertson
	Cellulose Acetate	Fibre	0.8	1.1	1.4	1.9	2.4	3.0	3.6	4.3	5.3	Robertson
Paper	M. F. Newsprint	Wood Pulp—24% Ash	2.1	3.2	4.0	4.7	5.3	6.1	7.2	8.7	10.6	U. S. B. of S.
	H. M. F. Writing	Wood Pulp—3% Ash	3.0	4.2	5.2	6.2	7.2	8.3	9.9	11.9	14.2	U. S. B. of S.
	White Bond	Rag—1% Ash	2.4	3.7	4.7	5.5	6.5	7.5	8.8	10.8	13.2	U. S. B. of S.
	Com. Ledger	75% Rag—1% Ash	3.2	4.2	5.0	5.6	6.2	6.9	8.1	10.3	13.9	U. S. B. of S.
	Kraft Wrapping	Coniferous	3.2	4.6	5.7	6.6	7.6	8.9	10.5	12.6	14.9	U. S. B. of S
Misc. Organic Materials	Leather	Sole Oak—Tanned	5.0	8.5	11.2	13.6	16.0	18.3	20.6	24.0	29.2	Phelps
	Catgut	Racquet Strings	4.6	7.2	8.6	10.2	12.0	14.3	17.3	19.8	21.7	Fuwa
	Glue	Hide	3.4	4.8	5.8	6.6	7.6	9.0	10.7	11.8	12.5	Fuwa
	Rubber	Solid Tire	0.11	0.21	0.32	0.44	0.54	0.66	0.76	0.88	0.99	Fuwa
	Wood	Timber (Av.)	3.0	4.4	5.9	7.6	9.3	11.3	14.0	17.5	22.0	Forest P. Lab.
	Soap	White	1.9	3.8	5.7	7.6	10.0	12.9	16.1	19.8	23.8	Fuwa
	Tobacco	Cigarette	5.4	8.6	11.0	13.3	16.0	19.5	25.0	33.5	50.0	Ford
Food-stuffs	White Bread		0.5	1.7	3.1	4.5	6.2	8.5	11.1	14.5	19.0	Atkinson
	Crackers		2.1	2.8	3.3	3.9	5.0	6.5	8.3	10.9	14.9	Atkinson
	Macaroni		5.1	7.4	8.8	10.2	11.7	13.7	16.2	19.0	22.1	Atkinson
	Flour		2.6	4.1	5.3	6.5	8.0	9.9	12.4	15.4	19.1	Bailey
	Starch		2.2	3.8	5.2	6.4	7.4	8.3	9.2	10.6	12.7	Atkinson
	Gelatin		0.7	1.6	2.8	3.8	4.9	6.1	7.6	9.3	11.4	Atkinson
Misc. Inorganic Materials	Asbestos Fibre	Finely Div.	0.16	0.24	0.26	0.32	0.41	0.51	0.62	0.73	0.84	Fuwa
	Silica Gel.		5.7	9.8	12.7	15.2	17.2	18.8	20.2	21.5	22.6	Fuwa
	Domestic Coke		0.20	0.40	0.61	0.81	1.03	1.24	1.46	1.67	1.89	Selvig
	Activated Charcoal	Steam Activated	7.1	14.3	22.8	26.2	28.3	29.2	30.0	31.1	32.7	Fuwa
	Sulphuric Acid	H_2SO_4	33.0	41.0	47.5	52.5	57.0	61.5	67.0	73.5	82.5	Mason

foodstuffs. Where the physical properties of the product affect value, the question of moisture is of special importance. With increase in moisture content, hygroscopic materials ordinarily become softer and more pliable. Economy of manufacturing, therefore, requires that the moisture content be maintained at a percentage most favorable to rapid and satisfactory manipulation and to a minimum loss of material through breakage. A constant condition is desirable in order that high speed machinery may be adjusted permanently for the desired production with a minimum loss from delays, wastage of raw material and defective product.

In the processing of hygroscopic materials, it is usually necessary to secure a final moisture content suitable for the goods as shipped. Where the goods are sold by weight it is proper that they contain a normal or standard moisture content. Air conditioning is important in certain branches of the chemical industry in controlling the temperature of reaction and facilitating or retarding evaporation. The control of moisture content of air supplied to blast furnaces in the manufacture of pig iron also has proved advantageous.

The moisture content of an hygroscopic material at any time depends upon the nature of the material and upon the temperature and especially the relative humidity of the air to which it has been exposed. Not only do different materials acquire different percentages of moisture after prolonged exposure to a given atmosphere, but the rate of absorption or drying out varies with the nature of the material, its thickness and density.

Table 1 shows the regain or hygroscopic moisture content of several organic and inorganic materials when in equilibrium at a dry-bulb temperature of 75 F and various relative humidities. The effect of relative humidity on regain of hygroscopic substances is clearly indicated. The effect of temperature is comparatively unimportant. In the case of cotton, for instance, an increase in temperature of 10 deg has the same effect on regain as a decrease in relative humidity of one per cent. Changes in temperature do, however, affect the rate of absorption or drying. Sudden changes in temperature cause temporary fluctuations in regain even when the relative humidity remains stationary.

Conditioning and Drying

Exposure of hygroscopic materials to an atmosphere of controlled humidity and temperature for the purpose of establishing a specified moisture condition in the material is called *conditioning*. Where the desired final moisture content is relatively low, the term *drying* is usually used. In any case, control of relative humidity, temperature, air velocity and length of exposure are all of more or less importance.

The conditioning treatment may be undertaken in a special enclosure (conditioning room) or it may be accomplished in the same room and at the same time as some regular manufacturing process. For instance, in the weaving of textiles a high relative humidity is commonly employed to keep the yarn strong and pliable, thus assisting in the weaving process and at the same time leaving the product in a satisfactory condition of regain for commercial reasons.

As a rule, commercial regain standards are specified percentages which by test have been found equivalent to a so-called *standard atmosphere* to

which the goods would be in hygroscopic equilibrium after prolonged exposure. Committee D13 on Textiles of the *American Society for Testing Materials* has adopted a relative humidity of 64 to 66 per cent and a temperature of 70 to 80 F as the standard atmosphere for textile testing.

ATMOSPHERIC CONDITIONS REQUIRED

The most desirable relative humidity during processing depends upon the product and the nature of the process. As far as the behavior of the material itself and its desired final condition are concerned, each material and process represents a different problem. The best relative humidity may range up to 100 per cent. Similarly the most desirable temperature may range between wide limits for different materials and treatments. Extremes in either relative humidity or temperature require relatively expensive equipment for maintaining these conditions and controlling them automatically. Also, in departments where people are working, their health, comfort and productive efficiency must be considered. A compromise often is desirable.

It is generally considered that relative humidities below 40 per cent are on the dry side, conducive to low regains, a brittle condition of fibrous materials, prevalence of static electricity and a tendency toward dryness of the skin and membranes of human beings. At the other end of the scale, humidities above 80 per cent are relatively damp, conducive to high regains, extreme softness and pliability.

Table 2 lists desirable temperatures and humidities for industrial processing. In using this table, care must be taken in qualifying the process. In preparing many materials, conditions are not maintained constantly, but different temperatures and humidities are held for varying lengths of time.

AIR CONDITIONING OF LIBRARIES[1]

Temperature has little effect on the preservation of books. A temperature over 100 F, combined with low relative humidity, may cause the book materials to become brittle, while a temperature much below freezing may cause permanent deterioration of the glue in the binding. The relative humidity should be maintained between 40 and 70 per cent, although these limits need not hold for short periods of time. If the relative humidity gets much below 40 per cent, first the glue and then the paper will tend to become brittle which will not cause any permanent damage unless the book is used while in this condition, as a subsequent increase in humidity will bring the materials back to their normal condition. If the relative humidity gets above 80 per cent, the growth of mildew may be expected.

One of the principal agents of destruction and deterioration of paper and books in libraries is sulphur dioxide gas in the air. If air containing sulphur dioxide is allowed to come in contact with cellulose, the principal constituent of paper, sulphuric acid is formed on the surface. This acid is not volatile at ordinary temperatures and therefore accumulates throughout the life of the paper. The destructive effect of the acid on the

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[1]See *U. S. Bureau of Standards Bulletin No.* 128 entitled, A Survey of Storage Conditions in Libraries, by Kimberly and Hicks.

TABLE 2. DESIRABLE TEMPERATURES AND HUMIDITIES FOR INDUSTRIAL PROCESSING

INDUSTRY	PROCESS	TEMPERATURE DEGREES FAHRENHEIT	RELATIVE HUMIDITY PER CENT
AUTOMOBILE	Assembly line	65	40
BAKING	Cake icing	70	50
	Cake mixing	75	65
	Dough fermentation room	80	76 to 80
	Loaf cooling	70	60 to 70
	Make-up room	75 to 80	55 to 70
	Mixing room	75 to 80	55 to 70
	Paraffin paper wrapping	80	55
	Proof boxes	80 to 90	80 to 95
	Storage of flour	70 to 80	60
	Storage of yeast	28 to 40	60 to 75
BIOLOGICAL PRODUCTS	Vaccines	below 32	
	Antitoxins	38 to 42	
BREWING	Fermentation in vat room	44 to 50	50
	Storage of grains	60	30 to 45
CERAMIC	Drying of auger machine brick	180 to 200	
	Drying of refractory shapes	110 to 150	50 to 60
	Molding room	80	60
	Storage of clay	60	35
CHEMICAL	General storage	60 to 80	35 to 50
CONFECTIONERY	Chewing gum rolling	75	50
	Chewing gum wrapping	70	45
	Chocolate covering	62 to 65	50 to 55
	Hard candy making	70 to 80	30 to 50
	Packing	65	50
	Starch room	75 to 85	50
	Storage	60 to 68	50 to 65
DISTILLERY	General manufacture	60	45
	Storage of grains	60	30 to 45
DRUG	Storage of powders and tablets	70 to 80	30 to 35
ELECTRICAL	Insulation winding	104	5
	Manufacture of cotton covered wire	60 to 80	60 to 70
	Manufacture of electrical windings	60 to 80	35 to 50
	Storage of electrical goods	60 to 80	35 to 50
FOOD	Butter making	60	60
	Dairy chill room	40	60
	Preparation of cereals	60 to 70	38
	Preparation of macaroni	70 to 80	38
	Ripening of meats	40	80
	Slicing of bacon	60	45
	Storage of apples	31 to 34	75 to 85
	Storage of bananas (See discussion p. 53)		
	Storage of citrus fruit	32	80
	Storage of eggs in shell	30	80
	Storage of meats	0 to 10	50
	Storage of sugar	80	35

Table 2. Desirable Temperatures and Humidities for Industrial Processing (Continued)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Fur	Drying of furs	110	
	Storage of furs	28 to 40	25 to 40
Hospital	Operating Room	75 to 80	50 to 70
Incubators	Chicken	99 to 102	55 to 75
	Human baby[a]		
	Room	75 to 88	59 to 65
	Incubator	76 to 100	59 to 65
Laboratory	General analytical and physical	60 to 70	60 to 70
	Storage of materials	60 to 70	35 to 50
Leather	Drying of hides	90	
Library	Bookstorage (see discussion in this chapter)	65 to 70	38 to 50
Linoleum	Printing	80	40
Lumber	Drying	180	30
Match	Manufacturing	72 to 74	50
	Storage of matches	60	
Munitions	Fuse loading	70	55
Paint	Drying of lacquers	60 to 80	25 to 50
	Drying of oil paints	60 to 90	25 to 50
	Brush and spray painting	60 to 80	25 to 50
Paper	Binding, cutting, drying, folding, gluing	60 to 80	25 to 50
	Storage of paper	60 to 80	35 to 45
Photographic	Development of film	70 to 75	60
	Drying	75 to 80	50
	Printing	70	70
	Cutting	72	65
Printing	Binding	70	45
	Folding	77	65
	Press room (general)	75	60 to 78
	Press room (lithographic)	60 to 75	20 to 60
	Storage of rollers	60 to 80	35 to 45
Rubber	Manufacturing	90	
	Dipping of surgical rubber articles	75 to 80	25 to 30
	Standard laboratory tests	80 to 84	42 to 48
Soap	Drying	110	70
Textile	Cotton— carding	75 to 80	50
	combing	75 to 80	60 to 65
	roving	75 to 80	50 to 60
	spinning	60 to 80	60 to 70
	weaving	68 to 75	70 to 80

[a]Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker, and K. D. Blackfan (A.S.H.V.E. Transactions, Vol. 36, 1930).

Table 2. Desirable Temperatures and Humidities for Industrial Processing (Continued)

Industry	Process	Temperature Degrees Fahrenheit	Relative Humidity Per Cent
Textile	Rayon— spinning	70	85
	twisting	70	65
	Silk— dressing	75 to 80	60 to 65
	spinning	75 to 80	65 to 70
	throwing	75 to 80	65 to 70
	weaving	75 to 80	60 to 70
	Wool— carding	75 to 80	65 to 70
	spinning	75 to 80	55 to 60
	weaving	75 to 80	50 to 55
Tobacco	Cigar and cigarette making	70 to 75	55 to 65
	Softening	90	85
	Stemming or stripping	75 to 85	70

paper is independent of the relative humidity of the surrounding air. Low alkaline concentration spray water may be used in an air washer to neutralize the acid condition. Such an air washer must be especially constructed to resist corrosion.

BANANA RIPENING

Ripe bananas are very perishable and for this reason men who deal in them must depend mainly upon control of the ripening speed as a means of regulating their daily supply of the fruit. Knowledge and experience are required in regulating the ripening treatment and to control the ripening speed. An accurate appraisal must be based upon a careful examination of the fruit when received to determine its condition, and periodically, thereafter, to determine the rate of ripening.

Fast ripening may be accomplished in from three to four days after the green fruit is placed in a ripening room by adjusting the temperatures of the room until the pulp temperature reaches about 70 F. In warming up cool fruit, quick heating is recommended, and it is good practice to use sufficient heat to raise the average fruit temperature at the rate of 2 to 3 deg per hour. After the first 24 hours, the room should be held at 68 F until the fruit is colored and then reduced to 66 F and held at this temperature. A high relative humidity of from 90 to 95 per cent should be maintained until the bananas show color when it may be reduced to about 80 per cent. High humidity is important during the warming period. No ventilation should be used until the fruit has colored, after which, ventilation at a rate not to exceed four changes per hour may be used to assist in reducing the humidity and to freshen the air in the room. If the fruit shows slow or uneven ripening characteristics, one or two applications of ethylene gas of approximately 1 cu ft per 1000 cu ft of room space may be used.

Medium speed ripening of bananas in from five to seven days may be accomplished by holding the fruit at 64 F. The humidity and ventilation control should be the same as for fast ripening. A treatment with ethylene gas will seldom be necessary. For slow ripening in from nine to ten

days, the fruit should be held at from 60 to 62 F. Temperatures below 62 F are not advisable for very thin fruit. The humidity should be the same as for fast ripening, and ventilation (up to 3 or 4 air changes per hour) should be used provided the humidity can be maintained. Ethylene gas treatment will not be required.

For holding ripened bananas, temperatures between 56 and 60 F are recommended. A reduction in humidity is beneficial in toughening the peel and reducing the mould, but too low a humidity will cause shrinkage. Although exact humidity control is not essential, the desirable range is between 75 and 80 per cent.

GREENHOUSES

Table 3 lists customary dry-bulb temperature ranges for different types of plants and flowers raised in greenhouses.

TABLE 3. CUSTOMARY TEMPERATURES FOR DIFFERENT TYPES OF GREENHOUSES

TYPE OF HOUSE	TEMPERATURE RANGE DEG FAHR	TYPE OF HOUSE	TEMPERATURE RANGE DEG FAHR
Carnation	45 to 55	Orchid, cool	50 to 55
Conservatory (general collection, winter garden, etc.)	60 to 65	Palm, warm	60 to 65
		Palm, cool	50 to 55
Cool	45 to 50	Propagating	55 to 60
Cucumber	65 to 70	Rose	55 to 60
Fern	60 to 65	Sweet pea	45 to 50
Forcing	60 to 65	Tomato	65 to 70
General purpose	55 to 60	Tropical	65 to 70
Lettuce	40 to 45	Violet	40 to 45
Orchid, warm	65 to 70		

APPARATUS FOR INDUSTRIAL CONDITIONING

Apparatus for industrial air conditioning may be divided into two distinct groups, namely, (1) *humidifiers* for increasing the moisture content of the air and for producing cooling by evaporation and (2) *dehumidifiers* for removing moisture from the air and for producing cooling by contact with water or surfaces at a lower temperature than the air.

Strictly speaking, humidity control alone, whether it involves humidification or dehumidification, is not *air conditioning*. To be entitled to this classification according to the definition in Chapter 42, the process should include the *simultaneous control* of temperature, humidity and air motion.

Industrial humidifiers may be divided into the following general types, according to the method of operation:

1. *Direct*, which spray into the room.
2. *Indirect*, which introduce moistened air.
3. *Combined* direct and indirect.

Spray Generation

Spray generation is obtained by (1) atomization, (2) impact, (3) hydraulic separation, and (4) mechanical separation.

Atomization involves the use of a compressed air jet to reduce the water particles to a fine spray. With the *impact* method, a jet of water under pressure impinges directly on the end of a small round wire. Where *hydraulic separation* is employed, a jet of water enters a cylindrical chamber and escapes through an axial port with a rapid rotation which causes it immediately to separate in a fine cone-shaped spray. In the *mechanical separation* process, water is thrown by centrifugal force from the surface of a rapidly revolving disc and separates into particles sufficiently small to be utilized in certain types of mechanical humidifiers.

Spray Distribution

Spray distribution is obtained by (1) air jet, (2) induction, and (3) fan propulsion.

The air jet which generates the spray in atomizers also carries the spray through a space sufficient for its distribution and evaporation, and this method of distribution is termed *air jet*. Where distribution is obtained by *induction*, the aspirating effect of an impact or centrifugal spray jet is utilized to induce a current of air to flow through a duct or casing and this air current distributes the spray. *Fan propulsion* obviously consists of the utilization of fans to entrain and distribute the spray.

Industrial type direct humidifiers are commonly classified as (1) atomizing, (2) high-duty, (3) spray and (4) self-contained or centrifugal.

Atomizing Humidifiers

There are several types of atomizing humidifiers, all of which rely upon compressed air as the atomizing and distributing agency, similar to the familiar method used in ordinary nasal atomizers. Compressed air (ordinarily about 30 lb per square inch) is supplied from a centrally-located air compressor through pipe lines to the atomizing units. The air lines are usually horizontal and parallel to water lines which supply water by gravity from a float tank. The water in the tank is maintained at a constant level slightly lower than the outlets of the atomizers themselves and is drawn constantly to the atomizer by aspiration when compressed air is supplied. This aspiration ceases and the flow of water stops when the air supply is cut off. The water should not be supplied under pressure to atomizers because of the possibility of leakage, drip, or coarse spray which cannot be permitted when water is supplied by aspiration.

High-Duty Humidifiers

Water is supplied to high-duty humidifiers under high pressure (usually about 150 lb per square inch) through pipe lines from a centrally-located pumping unit. The spray-generating nozzle which is of the impact type is located in a cylindrical casing. A drainage pan provides for the collection and return of unevaporated water which flows through a return pipe to a filter tank, from which it is recirculated. A powerful air current is forced through the humidifier by means of a fan mounted above the unit.

The air enters from above, is drawn through the head, charged with moisture, and cooled to the wet-bulb temperature. It then escapes from the opening below at a high velocity in a complete and nearly horizontal circle. The spray is quickly evaporated and the resulting vapor is rapidly

and thoroughly diffused. This effective distribution of fine spray over the maximum possible area insures complete and extremely rapid vaporization even at the highest humidities.

Spray Humidifiers

This type of humidifier consists of an impact spray nozzle in a cylindrical casing with a drainage pan below it. The aspirating effect of the spray nozzle induces a moderate air current through the casing which distributes the entrained spray. The general method of circulating and returning the water is similar to that employed for high-duty humidifiers. A suitable pump and centrally-located filter tank are required.

The spray and high-duty types of humidifiers have many features in common but the latter, because of its finer spray and greater capacity, is often considered better adapted for producing high humidities.

Self-Contained Humidifiers

The self-contained or centrifugal humidifier has the ability to generate and distribute spray without the use of air compressors, pumps, or other auxiliaries and is therefore convenient for small installations where few heads are required. These humidifiers are generally of two types, one of which operates under low water pressure and evaporates about 20 per cent of the water fed to it, while the other maintains a constant level in the bottom of the humidifier and may be arranged to evaporate all the water supplied to it.

Humidifiers and air washers are also described in Chapter 11.

Where large quantities of power are generated in a limited space and where a comparatively high relative humidity is required, it is often feasible and economical to use a combination of direct and indirect humidification. The indirect humidification provides the desired quantity of ventilation and cooling, and the additional direct humidification provides for increase in humidity without interfering with the ventilation or the cooling effected by the indirect system.

In general, it may be stated that direct humidification is most satisfactory where high humidities are desired but where little cooling, ventilation or air motion is required. Therefore, the indirect system is most applicable where either low or high relative humidities are desired with maximum cooling and ventilation effect. For conditions that require an unusually large amount of heat to be absorbed by ventilation, together with the maintenance of high humidities, it is often preferable to make use of the combination system of indirect and direct humidification. If the indirect system alone were used it would mean an unusually large volume of air to be handled, which might interfere, due to air motion, with production, even though it would result in greater cooling effect. If direct humidification alone were used, no ventilation would be obtained, with consequently higher room temperatures.

Dehumidifiers, which are similar in design and appearance to indirect humidifiers and air washers, are described in Chapter 11. The main differences are found in the internal construction of the dehumidifier, in the use of refrigeration or of heat as required for controlling the water temperature, and in differences in the general methods of control.

Chapter 4

NATURAL VENTILATION

Wind Forces, Stack Effect, Openings, Windows, Doors, Skylights, Roof Ventilators, Stacks, Principles of Control, General Rules, Measurements, Dairy Barn Ventilation, Garage Ventilation

VENTILATION by natural forces, supplemented in certain cases with mechanical forces, finds extensive application in industrial plants, public buildings, schools, dwellings, garages, and in farm buildings.

NATURAL VENTILATION FORCES

The natural forces available for the displacement of air in buildings are the wind and the difference in temperature of the air inside and outside the building. The arrangement and control of ventilating openings should be such that the two forces act cooperatively and not in opposition.

Wind Forces

In considering the use of natural wind forces for the operation of a ventilating system, account must be taken of (1) average and minimum wind velocities, (2) wind direction, (3) seasonal, daily and hourly variations in wind velocity and direction, and (4) local wind interference by buildings and trees.

Table 1, Chapter 8, gives values for the average summer wind velocities and the prevailing wind directions in various localities throughout the United States, while Table 2, Chapter 7, lists similar values for the winter. In almost all localities the summer wind velocities are lower than those in the winter, and in about two-thirds of the localities the prevailing direction is different during the summer and winter. While average wind velocities are seldom below 5 mph, there are many hours in each month during which the wind velocity is from 3 to 5 mph, even in localities where the seasonal average is considerably above 5 mph. There are relatively few places where the hourly wind velocity falls much below 3 mph for more than 10 daylight hours per month. Usually a natural ventilating system should be designed to operate satisfactorily with a wind velocity of 3 to 6 mph, depending on locality.

The following formula may be used for calculating the quantity of air forced through ventilation openings by the wind, or for determining the proper size of such openings:

$$Q = EAV \tag{1}$$

where

Q = air flow in cubic feet per minute.

A = free area of inlet (or outlet) openings in square feet.
V = wind velocity in feet per minute.
= miles per hour × 88.
E = effectiveness of openings.

(E should be taken at from 50 to 60 per cent if the inlet openings face the wind and from 25 to 35 per cent if the inlet openings receive the wind at an angle).

If outlet openings, where air leaves a building, are smaller than inlet openings, where air enters a building, the air will be less effective than indicated by the constant E.

The accuracy of the results obtained by the use of Formula 1 depends upon the placing of the openings, as the formula assumes that ventilating openings have a flow coefficient slightly greater than that of a square-edge orifice. If the openings are not advantageously placed with respect to the wind, the flow per unit area of the openings will be less, and if unusually well placed, the flow will be slightly more than that given by the formula. Inlets should be placed to face directly into the prevailing wind, while outlets should be placed in one of the following four places:

1. On the side of the building directly opposite the direction of the prevailing wind.
2. On the roof in the low pressure area caused by the jump of the wind (see Fig. 1).
3. In a monitor on the side opposite from the wind.
4. In roof ventilators or stacks exposed to the full force of the wind[1].

Forces due to Stack Effect[2]

The stack effect produced within a building is due to the difference in weight of the warm column of air within the building and the cooler air outside. The flow due to stack effect is proportional to the square root of the draft head, or approximately:

$$Q = 9.4\,A\,\sqrt{H\,(t_1 - t_2)} \qquad (2)$$

where

Q = air flow in cubic feet per minute.
A = free area of inlets or outlets (assumed equal) in square feet.
H = height from inlets to outlets, in feet.
t_1 = *average* temperature of indoor air in height H, in degrees Fahrenheit.
t_2 = temperature of outdoor air, in degrees Fahrenheit.
9.4 = constant of proportionality, including a value of 65 per cent for effectiveness of openings. This should be reduced to 50 per cent (constant = 7.2) if conditions are not favorable.

The height between inlets and outlets should be the maximum which the building construction will allow.

In some cases the necessary air flow will be known from the requirements of the building occupancy, and the area necessary for certain assumed temperature differences may be calculated. Or the areas may be fixed by the building construction, and the maximum air flow for various differences between indoor and outdoor temperatures may be

[1]See Airation of Industrial Buildings, by W. C. Randall (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).
[2]See Neutral Zone in Ventilation, by J. E. Emswiler (A.S.H.V.E. TRANSACTIONS, Vol. 32, 1926), and Predetermining Airation of Industrial Buildings, by W. C. Randall and E. W. Conover (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

calculated. In any case, the conditions which give the minimum air flow are those which control the design, as the system must have ample capacity even under the most unfavorable conditions which are those of mild or warm weather.

OPENINGS FOR NATURAL VENTILATION

The engineering problems of a natural ventilation system consist in the

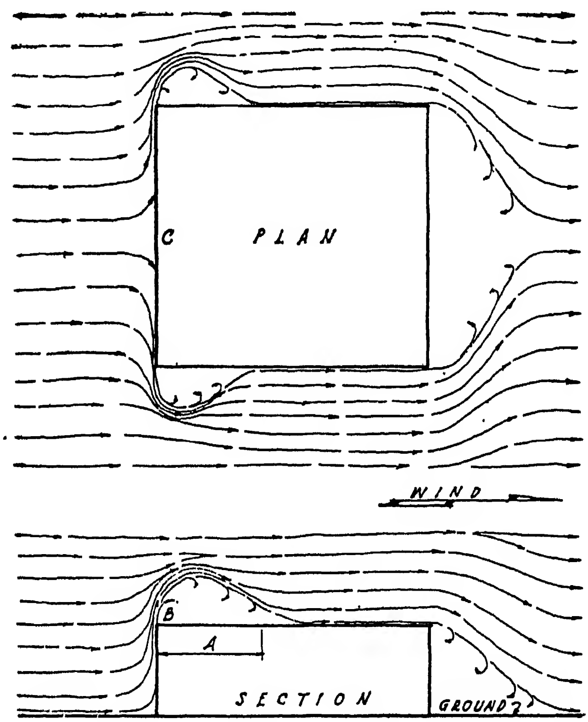

FIG. 1. THE JUMP OF WIND FROM WINDWARD FACE OF BUILDING. (*A*—LENGTH OF SUCTION AREA; *B*—POINT OF MAXIMUM INTENSITY OF SUCTION; *C*—POINT OF MAXIMUM PRESSURE)

design, location, and control of ventilating openings to best utilize the natural ventilation forces, in accordance with the requirements of building occupancy. The types of openings may be classified as:

1. Windows, doors, monitor openings, and skylights.
2. Roof ventilators.
3. Stacks connecting to registers.
4. Specially designed inlet or outlet openings.

Windows, Doors and Skylights

Windows have the advantage of transmitting light, as well as providing ventilating area when open. Their movable parts are arranged to open in

various ways; they may open by sliding as in the ordinary double-hung windows, by tilting on horizontal pivots at or near the center, or by swinging on pivots at the top or bottom. Whatever the form and type of window used, the amount of clear area that can be made available is the factor of greatest importance in ventilation.

All types of sash (double-hung, top, center or bottom horizontal pivoted, or vertical pivoted) have about the same air flow capacity for the same clear area. Air leakage through *closed* windows is important during high winds (Chapter 6).

The proper distribution of air in occupied spaces is an element almost as important as that of sufficient air quantity. Advantageous pivoting of sash is very useful for securing good air distribution. Deflectors are sometimes used for the same purpose, and these devices should be considered a part of the ventilation system.

Door openings are seldom included in the ventilation calculations, though they may be of great value for extreme summer conditions, and should be considered in this connection.

Skylight and monitor openings are of importance as these and the roof ventilators are outlets, while the lower windows are usually inlets on the windward side and outlets on the leeward side. In general the areas of inlets and of outlets should be about equal. It is important to make a check on this ratio in any installation, as any great excess of area of one set of openings over another means waste opening area. The operating devices used for sash, monitors, skylights and roof ventilators should be well selected as poor operating devices may defeat the entire design.

Roof Ventilators

The function of a roof ventilator is to provide a storm and weather proof air outlet, which is sensitive to wind action for producing additional flow capacity, and at the same time is subject to manual or automatic control by suitable dampers. The capacity of a ventilator at a constant wind velocity and temperature difference, depends upon four things: (1) its location on the roof, (2) the resistance it offers to air flow, (3) the area and location of openings provided for air inflow at a lower level, and (4) the ability of the ventilator head to utilize the kinetic energy of the wind for inducing flow by centrifugal or ejector action. Frequently one or more of these capacity factors is overlooked in a ventilator installation.

For maximum flow induction, a ventilator should be located on that part of the roof which receives the full wind without interference. (See Fig. 1). This does not mean that no ventilators are to be installed within the suction region created by the wind jumping over the building, or in a light court, or on a low building between two high buildings. Ventilators are highly effective in such low-pressure areas, but their ejector action, caused by wind velocity, is of little importance in these locations, and hence their size should be increased proportionally.

Ventilator resistance depends on (1) type of inlet, (2) area of openings and passages and (3) number of turns or changes of direction of the air flow. The inlet grille, if any, should have ample free area, and the ventilator should always be provided with a taper-cone inlet in order to produce the effect of a bell-mouth nozzle (flow coefficient 0.97) rather than that of

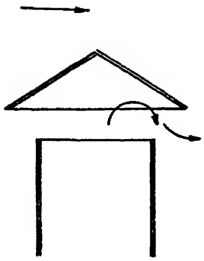

FIG. 2

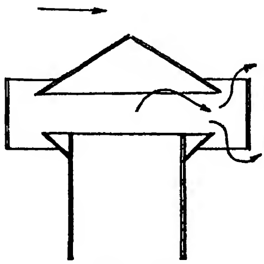

FIG. 3

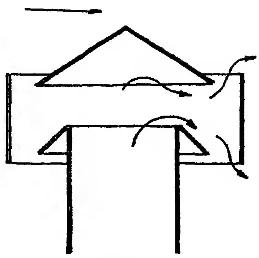

FIG. 4

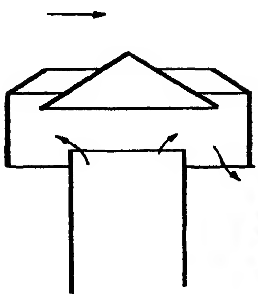

FIG. 5

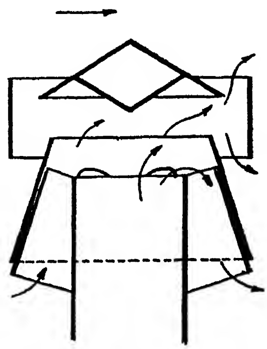

FIG. 6

TYPES OF STATIONARY VENTILATORS

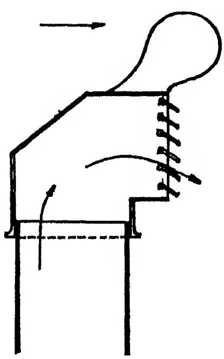

FIG. 7

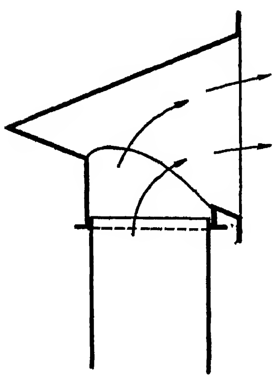

FIG. 8

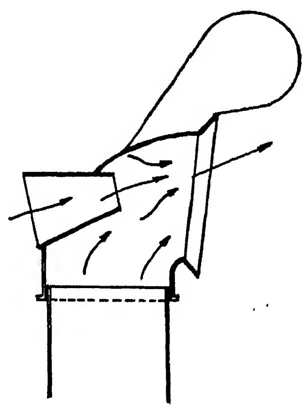

FIG. 9

TYPES OF OSCILLATING VENTILATORS

a square-entrance orifice (flow coefficient 0.60). In other words, the grille should be oversize as compared with the ventilator, and they should be connected by a tapering collar. If the ventilator head construction produces changes in the direction of air flow, the area of the flow passages should be increased accordingly.

Air inlet openings at lower levels in the building are of course necessary for the economical use of ventilator capacity. The inlet openings should be at least equal to, and preferably twice as great as the combined throat areas of all roof ventilators. The air discharged by a roof ventilator depends on wind velocity and temperature difference, but due to the four capacity factors already mentioned, no simple formula can be devised for expressing ventilator capacity.

Several types of roof ventilators are shown in Figs. 2 to 11. These may be classified as *stationary*, Figs. 2 to 6, *pivoted* or *oscillating*, Figs. 7 to 9, or *rotating*, Figs. 10 and 11. When selecting unit ventilators, some attention should be paid to ruggedness of construction, storm-proofing features, dampers and damper operating mechanisms, possibilities of noise from dampers or other moving parts, and possible maintenance costs.

It should be kept in mind that a suitable combination of roof ventilators with mechanical ventilation frequently offers the best solution of a ventilating problem. The natural ventilation units may be used to supplement power driven supply fans, and under favorable weather conditions it may be possible to shut down the power driven units. Where low operating costs are very important, such a combination has great advantages. Roof ventilators with built-in electric fans are attracting increased attention because they combine the advantages of low installation and operating cost with those of continuous service.

Controls

In connection with any combination between natural and fan ventilation, the controls are of importance. Both the fans and the ventilator dampers may be controlled by some combination of three methods: (1) hand operation, (2) thermostat operation, and (3) control by wind velocity. The thermostat station may be located anywhere in the building, or it may be located within the ventilator itself. The purpose of wind velocity control is to obtain a definite volume of exhaust regardless of the natural forces, the fan motor being energized when the natural exhaust capacity falls below a certain minimum, and again shut off when the wind velocity rises to the point where this minimum volume can be supplied by natural forces.

Stacks

Stacks are really chimneys and utilize both the inductive effect of the wind and the force of temperature difference (the so-called *gravity* action). While their openings projecting above the roof are not provided with any special construction for developing suction by the action of the wind, the plain vertical opening is also effective in this respect. Like the roof ventilator, the stack outlet should be located so that the wind may act upon it from any direction.

Stacks are applicable particularly in the case of schools, apartments, residences and small office buildings. Partitions interfere with general air circulation, and some type of outlet from each room is necessary. If the building is not too tall, and the requirements of occupancy are moderate, a system of stacks with registers in each room may be more economical then a system of mechanical ventilation employing fans. In making the comparison, however, the building space occupied by the stacks should be considered.

With little or no wind, chimney effect or temperature difference will produce outflow through the stacks and an equal inflow through windows in all sides of the building. With wind, the inductive force at the top of ventilating shafts is more powerful than that on the leeward side of the

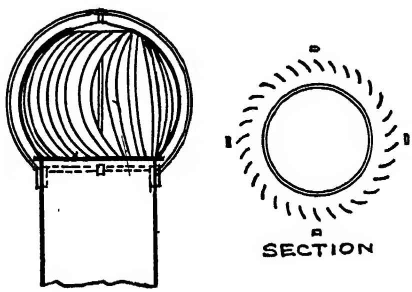

FIG. 10. ROTATING VENTILATOR

building, so that air is drawn in through leeward openings by a combination of the forces of wind and temperature difference. On the windward side, the direct forcing pressure of the wind is of course added to the temperature difference effect. Thus forces are available for causing inflow at practically every window of such a building. Adequacy of stack size must, of course, be provided.

PRINCIPLES OF AIR FLOW CONTROL

The air flow through a ventilation opening depends on the two factors already discussed, namely, (1) the natural forces available, (2) the openings available, and the resistance to flow offered by these openings. The design problem includes, of course, a determination of the *desired air quantity and distribution* in order that the openings may be properly placed.

The purpose of ventilation is to carry off either excess heat or air impurities, and the desired air quantities depend upon the amount of heat or of impurities present. The amount of heat can be determined, in the case of forge shops for example, from the amount of fuel burned, which in turn is based upon the production capacity for which the building is being designed. In the case of foundries, the heat given off by the metal in cooling from the molten state can be used. In some instances, not all

of the heat may be dissipated to the air, but a fair estimate of the amount to be removed by the air can usually be made.

The next step is to select the temperature difference to be maintained. Knowing the amount of heat to be removed and having selected a desirable temperature difference, the amount of air to be passed through the building per minute to maintain this temperature difference can be determined by means of the following equation:

$$H = \frac{c\,Q\,D}{V} \qquad (3)$$

where

c = 0.24 = specific heat of air.

V = specific volume of the air, cubic feet per pound, about 13.5. (See Chapter 41).

H = heat to be carried off, in Btu per minute.

Q = air flow in cubic feet per minute.

D = inlet-outlet temperature difference in degrees Fahrenheit.

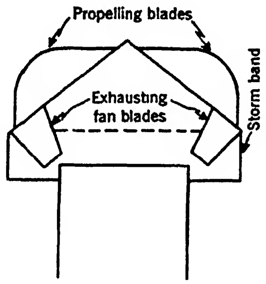

FIG. 11. ROTATING ROOF VENTILATOR

For disposing of air impurities, the required air flow must be such that the outside air will dilute the impurities to a degree that they are no longer objectionable. For human occupancy, such as in auditoriums and classrooms, 10 cfm per person is usually taken as the minimum of outside air necessary for ventilation (see Chapter 2). For garage ventilation, sufficient air must be admitted to dilute the carbon monoxide content of the indoor air to 1 in 10,000 (see Garage Ventilation in this Chapter).

Air *quantity* and *quality* are not the only requirements. For human occupancy, air *distribution* is important. In ventilation the air distribution is almost entirely a matter of the number, the design, and the location of inlets and outlets. In locating openings, special precautions should be taken against the formation of dead air spaces or *pockets* within the zone of occupancy.

Suggested methods for estimating the air flow due to temperature difference alone and to wind alone have already been given. It must be remembered that when both forces are acting together, even without interference, the resulting air flow is not equal to the sum of the two

estimated quantities. The same openings have been assumed in both cases, and since the resistance to flow through the openings varies approximately with the square of the velocity[3], this resistance becomes a limiting factor as the flow through the openings is increased.

Recent investigations[1 2] show that the total flow is only 10 per cent above the flow caused by the greater force when the two forces are nearly equal, and this percentage decreases rapidly as one force increases above the other. Tests on roof ventilators indicate that this is too conservative in the direction of low total flow quantities, but there is in any case a large judgment factor involved. The wind velocity and direction, the outdoor temperature, or the indoor activities cannot be predicated with certainty, and great refinement in calculations is therefore not justified. When designing for winter conditions, an added variable is the heat lost by direct flow through walls and windows and by infiltration.

Example 1. Assume a drop forge shop, 200 ft long, 100 ft wide, and 30 ft high. The cubical content is 600,000 cu ft, and the height of the air outlet over that of the inlet is 30 ft. Oil fuel of 18,000 Btu per lb is used in this shop at the rate of 15 gal per hour (7.75 lb per gal). Temperature differences are 10 deg Fahr in summer and 30 F in winter, and the wind velocity is 5 mph in summer and 8 mph in winter. What is the necessary area for the inlets and outlets, and what is the rate of air flow through the building?

Solution. The system must be designed for the summer conditions as these are the more severe. The heat to be removed per minute is:

$$H = \frac{15}{60} \times 7.75 \times 18{,}000 = 34{,}875 \text{ Btu per minute.}$$

By Equation 3, the air flow required to remove this heat with a temperature difference of 10 deg is:

$$Q = \frac{VH}{cD} = \frac{13.5 \times 34{,}875}{0.24 \times 10} = 196{,}172 \text{ cfm.}$$

This is equal to 19.6 air changes per hour. The assumption is made that the average temperature difference between indoors and outdoors is the same as the temperature rise of the air from the inlet opening to the outlet opening. Actually, the latter difference is larger and so the value of 19.6 air changes per hour is conservative as it allows for more cooling than is necessary for an *average* temperature difference of 10 deg.

If 196,172 cfm are to be circulated by the force of the temperature difference alone, the area of opening would be, by Equation 2:

$$A = \frac{Q}{9.4\sqrt{H\,(t_1 - t_2)}} = \frac{196{,}172}{9.4\sqrt{30 \times 10}} = 1{,}205 \text{ sq ft}$$

If this area of openings were provided, a wind velocity of 5 mph, acting alone, would produce a flow according to Equation 1, of:

$$Q = EAV = 0.50 \times 1{,}205 \times 5 \times 88 = 265{,}100 \text{ cfm.}$$

If the inlet openings do not face the wind, but are at an angle with it, about half this amount may be considered to flow.

A factor of judgment must now be exercised in making the selection of the area of openings to be specified. Apparently 1205 sq ft are a very

[3]This is true for *turbulent* flow only. It would be more correct to state that the resistance varies approximately with V^2 for high to moderate velocities, with $V^{1.8}$ for moderate to low velocities, and with the first power of the velocity for very low velocities through small openings.

generous allowance because either a direct wind of 5 mph or an average temperature difference of 10 deg acting alone will more than suffice to carry away the heat, and when the two forces are acting together, the system may have an excess capacity of 25 per cent to 50 per cent, especially if the outlets are made up partially of roof ventilators which employ the force of the wind for producing a suction effect. On the other hand, the wind may at times come from an unfavorable direction, or its velocity may fall below 5 mph or the building construction may not permit a full 2400 sq ft of inlet window area and an equal amount of monitor or roof ventilator outlet area. In case the two sets of openings are not equal, their effectiveness is reduced.

From this example it must be apparent that while formulas may furnish a reliable guide, the final solution of a problem of natural ventilation requires a common sense analysis of local conditions to supplement and to modify the dictates of the formulas.

GENERAL RULES

A few of the important requirements in addition to those already outlined are:

1. Inlet openings should be well distributed, and should be located on the windward side near the bottom, while outlet openings are located on the leeward side near the top. Outside air will then be supplied to the zone of occupancy.

2. Direct short circuits between openings on two sides at a high level may clear the air at that level without producing any appreciable ventilation at the level of occupancy.

3. Roof ventilators should be located 20 to 40 ft apart each way and preferably on the ridge of the roof. The closer spacings are used when ventilating rooms with low ceilings.

4. Greatest flow per square foot of total opening is obtained by using inlet and outlet openings of nearly equal areas.

5. In an industrial building where furnaces, that give off heat and fumes, are to be installed, it is better to locate them in the end of the building exposed to the prevailing wind. The strong suction effect of the wind at the roof near the windward end will then cooperate with temperature difference, to provide for the most active and satisfactory removal of the heat and gas laden air.

6. In case it is impossible to locate furnaces in the windward end, that part of the building in which they are to be located should be built higher than the rest, so that the wind, in splashing therefrom will create a suction. The additional height also increases the effect of temperature difference to cooperate with the wind.

7. In the use of monitors, windows on the windward side should usually be kept closed, since, if they are open, the inflow tendency of the wind counteracts the outflow tendency of temperature difference. Openings on the leeward side of the monitor result in cooperation of wind and temperature difference.

8. In order that the force of temperature difference may operate to maximum advantage, the vertical distance between inlet and outlet openings should be as great as possible. Openings in the vicinity of the neutral zone are less effective for ventilation.

9. In order that temperature difference may produce a motive force, there must be vertical distance between openings. That is, if there are a number of openings available in a building, but all are at the same level, there will be no motive head produced by temperature difference, no matter how great that difference might be.

10. In the design of window ventilated buildings, where the direction of the wind is quite constant and dependable, the orientation of the building together with amount

and grouping of ventilation openings can be readily arranged to take full advantage of the force of the wind. On the other hand, where the direction of the wind is quite variable, it may be stated as a general principle that windows should be arranged in sidewalls and monitors so that there will be approximately equal area on all sides. Thus, no matter what the wind's direction, there will always be some openings directly exposed to the pressure force of the wind, and others opposed to a suction force, and effective movement through the building will be assured.

11. The intensity of suction or the vacuum produced by the jump of the wind is greatest just back of the building face. The area of suction does not vary with the wind velocity, but the flow due to suction is directly proportional to wind velocity.

12. Openings much larger than the calculated areas are sometimes desirable, especially when changes in occupancy are possible, or to provide for extremely hot days. In the former case, free openings should be located at the level of occupancy for psychological reasons.

13. Special consideration should be given to the possibility of sidewall or monitor windows being closed on account of weather conditions. Such possibilities favor roof ventilators and specially designed storm-proof inlets.

MEASUREMENT OF NATURAL VENTILATION

The determination of the performance of any ventilating system involves measurements which are not easy to make. The difficulties are increased in the case of natural ventilation, since the motive forces and the air velocities are very small. The measurements necessary for giving the *capacity* of a system are (1) velocity of the wind, (2) velocity of the air through inlet and outlet openings, (3) outdoor air temperature, and (4) average indoor air temperature.

Measuring Wind Velocity. The cup-type of anemometer as used for Weather Bureau observations is sufficiently accurate for this measurement. Some more accurate instruments as well as direct-reading types have been developed for airport service, but for ventilation work it is the average wind velocity over a long period which determines the capacity of the system. Hence the use of the Weather Bureau instrument, with an observation period of one hour or more, is satisfactory. If observations of wind direction are required, these should be taken by observing a sensitive weather vane at frequent intervals (about every 5 minutes) during the same period.

Velocity of Air Through Openings. The vane type anemometer is the most practical instrument for this measurement.

Use a small (4 in.) low-speed anemometer, and correct all readings according to a recent calibration. Mount the anemometer in a strap iron clamp with a long handle for convenience. Divide each opening into 5 in. squares (by string or wire) and hold the anemometer in the center of each square for a definite period of from 15 to 30 seconds. Record the result of the traverse as soon as completed and start another one immediately. A series of traverses over a period of one hour, or the full period covered by the wind velocity observations with a fairly steady wind, may be considered a satisfactory test for that wind velocity. It is preferable to have an anemometer observer at each opening. If the opening is covered by a grille or register, use the proper correction factors (see Chapter 40).

Outdoor Temperature. It is easy to make an error of 1 to 5 deg in

observing the outdoor air temperature. An accurate thermometer, calibrated in 1 deg divisions should be used. The thermometer should be mounted in the shade at about mid-height of the building and not too near the building wall or adjacent to an air outlet. The heat from a wall or roof which has been exposed to the sun is easily transmitted to a thermometer, with resulting high readings.

Average Indoor Temperature. It is important to note that the capacity of an opening (such as roof ventilator) does *not* depend on the difference in the temperatures measured adjacent to the opening. It depends rather on the difference between the *average* temperature of the column of air inside the building and that outside. Indoor temperatures should therefore be observed at various heights to secure a good average.

DAIRY BARN VENTILATION[4]

A successful barn ventilating system is one which continuously supplies the proper amount of air required by the stock, with proper distribution and without drafts, and one which removes the excessive heat, moisture, and odors, and maintains the air at a proper temperature, relative humidity, and degree of cleanliness.

Barn temperatures below freezing and above 80 F affect milk production. Milk producing stock should be kept in a barn temperature between 45 and 50 F. Dry stock, at reduced feeding, may be kept in a barn 5 to 10 deg higher. Calf barns are generally kept at 60 F, while hospital and maternity barns usually have a temperature of 60 F or somewhat higher.

The heat produced by a cow of an average weight of 1000 lb may be taken as 3000 Btu per hour. The average rate of moisture production by a cow giving 20 lb of milk per day is 15 lb of water per day, or 4375 grains per hour. To set a standard of permissible relative humidity for cow barns is difficult. For 45 F an average relative humidity of 80 per cent is satisfactory, with 85 per cent as a limit.

Where the barn volume is within the limit that can be heated by the stabled animals, the air supply need not be heated. The air should be supplied through or near the ceiling. It is better to have the exhaust openings near the floor as larger volumes of warm air are then held in the barn and there is better temperature control with less likelihood of sudden change in barn temperature.

If a cow weighs 1000 lb and produces 3000 Btu of heat per hour, and if a barn for the cow has 600 cu ft of air space with 130 sq ft of building exposure, one cow will require 2600 to 3550 cfh of ventilation, depending on the temperature zone in which the barn is located. The permissible heat losses through the structure, based on one cow and depending on the temperature zone, vary between 0.043 and 0.066 Btu per hour per cu ft of barn space, and 0.197 to 0.305 Btu per hour per sq ft of barn exposure.

[4]For additional information on this subject refer to *Technical Bulletin*, U. S. Department of Agriculture (1930), by M. A. R. Kelley.

Dairy Barn Ventilation, by F. L. Fairbanks (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

Cow Barn Ventilation, by Alfred J. Offner (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, January, 1933).

GARAGE VENTILATION

On account of the hazards resulting from carbon monoxide and other physiologically harmful or combustible gases or vapors in garages, the importance of proper ventilation of these buildings cannot be overemphasized. During the warm months of the year, garages are usually ventilated adequately because the doors and windows are kept open. As cold weather sets in, more and more of the ventilation openings are closed and consequently on extremely cold days the carbon monoxide concentration runs high.

Many garages can be satisfactorily ventilated by natural means[5] particularly during the mild weather when doors and windows can be kept open. However, the A.S.H.V.E. Code for Heating and Ventilating Garages, adopted in 1929, states that natural ventilation may be employed for the ventilation of storage sections where it is practical to maintain open windows or other openings at all times. The code specifies that such openings shall be distributed as uniformly as possible in at least two outside walls, and that the total area of such openings shall be equivalent to at least 5 per cent of the floor area. The code further states that where it is impractical to operate such a system of natural ventilation, a mechanical system shall be used which shall provide for either the supply of 1 cu ft of air per minute from out of doors for each square foot of floor area, or for removing the same amount and discharging it to the outside as a means of flushing the garage.

Research

Research on garage ventilation undertaken by the A.S.H.V.E. Committee on Research at Washington University, St. Louis, Mo., and at the University of Kansas, Lawrence, Kans., in cooperation with the A.S.H.V.E. Research Laboratory, and at the A.S.H.V.E. Research Laboratory has resulted in authoritative papers[6] on the subject.

Some of the conclusions from work at the Laboratory are listed below:

1. Upward ventilation results in a lower concentration of carbon monoxide at the breathing line and a lower temperature above the breathing line than does downward ventilation, for the same rate of carbon monoxide production, air change and the same temperature at the 30-in. level.

2. A lower rate of air change and a smaller heating load are required with upward than with downward ventilation.

3. In the average case upward ventilation results in a lower concentration of carbon

[5]Code for Heating and Ventilating Garages (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

Airation Study of Garages, by W. C. Randall and L. W. Leonhard (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[6]Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

Carbon Monoxide Distribution in Relation to the Ventilation of an Underground Ramp Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Carbon Monoxide Distribution in Relation to the Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

Carbon Monoxide Distribution in Relation to the Heating and Ventilation of a One-Floor Garage, by F. C. Houghten and Paul McDermott (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, July, 1933).

Carbon Monoxide Surveys of Two Garages, by A. H. Sluss, E. K. Campbell and Louis M. Farber (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, December, 1933).

monoxide in the occupied portion of a garage than is had with complete mixing of the exhaust gases and the air supplied. However, the variations in concentration from point to point, together with the possible failure of the advantages of upward ventilation to accrue, suggest the basing of garage ventilation on complete mixing and an air change sufficient to dilute the exhaust gases to the allowable concentration of carbon monoxide.

4. The rate of carbon monoxide production by an idling car is shown to vary from 25 to 50 cfh, with an average rate of 35 cfh.

5. An air change of 350,000 cfh per idling car is required to keep the carbon monoxide concentration down to one part in 10,000 parts of air.

Chapter 5

HEAT TRANSMISSION

Heat Transfer Through Walls, Areas Where Transmission Losses Occur, Calculations for Transmission Losses, Coefficients of Transmission and Tables, Temperatures and Coefficients, Air Spaces, Surface Conductances, Conductivities of Materials

THIS chapter concerns the transmission losses of a building which, in conjunction with the infiltration losses, must always be considered in arriving at the size of the heating (or cooling) plant required for the maintenance of certain specified inside temperature conditions.

HEAT TRANSFER

Whenever a difference in temperature exists between the two sides of any structural material, such as a wall or roof of a building, a transfer of heat takes place through that material. When the inside temperature is the higher, heat reaches or enters the inside surface of the wall by radiation and convection, because the air and objects within the building are always warmer than the inside surface of the wall when the inside air temperature t is greater than the outside air temperature t_o. This heat must then pass through the material of the wall from the inside to the outside surface by conduction, and is finally given off from the outside surface by radiation and convection, provided, of course, that equilibrium has been established and all four temperatures are constant. If the outside temperature is the higher, the reverse process takes place.

CALCULATIONS FOR TRANSMISSION LOSSES

The calculations for heat transmission losses are made by multiplying the area A in square feet of wall, glass, roof, floor, or material through which the loss takes place, by the proper coefficient U for such construction or material and by the temperature difference between the inside air temperature t at the proper level (in many cases not the *breathing-line*) and the outside air temperature t_o. Therefore,

$$H_t = A\,U\,(t - t_o) \tag{1}$$

where

H_t = Btu per hour transmitted through the material of the wall, glass, roof or floor.

A = area in square feet of wall, glass, roof, floor, or material, taken from building plans or actually measured. (Use the net inside or heated surface dimensions in all cases).

$t - t_o$ = temperature difference between inside and outside air, in which t must always be taken at the proper level. Note that t may not be the *breathing-line* temperature in many cases.

AREAS WHERE TRANSMISSION LOSSES OCCUR

Heat is lost from a building by transmission through all of those surfaces which separate heated spaces from the outside air or from unheated colder spaces within the building. In general, five kinds of surfaces are involved: (1) outside walls; (2) outside glass; (3) inside walls or partitions next to unheated spaces; (4) ceilings of upper floors, either below a cold attic space or as the underside of a roof slab; and (5) floors of heated rooms above an unheated space.

The net outside wall surface is usually determined by reference to the scale plans and elevations of the building concerned. In some cases, of course, the actual building may have to be measured. The total area of all outside openings which are occupied by windows and doors is accurately measured and listed as *glass*. The *glass* area is then deducted from the total outside wall area for each room and the difference is the net wall area. The outside wall areas for any floor should be based on the vertical floor-to-floor heights and the horizontal distance from center to center of partitions separating different rooms. If there are no partitions, measure from inside face of one wall to inside face of next wall. The areas of walls, ceilings and floors next to cold or unheated spaces are found, of course, by taking the inside dimensions of such areas, measured on the heated side.

COEFFICIENTS OF TRANSMISSION

The coefficients of transmission may be determined by means of the guarded hot box or the Nicholls Heat Meter described in Chapter 40, or they may be calculated from fundamental constants. On account of the unlimited number of combinations of building materials, it would be impractical to attempt to determine by test the heat transmission coefficients of every type of construction in use; consequently, in most cases it is advisable to calculate these coefficients.

Symbols

The following symbols are used in the heat transmission formulae in this chapter:

U = Thermal transmittance or over-all coefficient of heat transmission and is the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 F between the air on the inside and outside of the wall, floor, roof or ceiling.

k = Thermal conductivity and is the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 F between the two surfaces of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

C = Thermal conductance and is the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a non-homogeneous material for the thickness or type under consideration for a difference in temperature of 1 F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plaster board and hollow clay tile.

f = Film or surface conductance and is the amount of heat expressed in Btu transmitted by radiation, conduction and convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature

of 1 deg between the surface and the surrounding air. To differentiate between inside and outside wall (or floor, roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance.

a = Thermal conductance of an air space and is the amount of heat expressed in Btu transmitted by radiation, conduction and convection in one hour through an area of 1 sq ft of an air space for a temperature difference of 1 F. The conductance of an air space depends on the mean absolute temperature, the width, the position and the character of the materials enclosing it.

R = Resistance or resistivity which is the reciprocal of transmission, conductance, or conductivity, *i.e.:*

$\frac{1}{U}$ = over-all or air-to-air resistance.

$\frac{1}{k}$ = internal resistivity.

$\frac{1}{C}$ = internal resistance.

$\frac{1}{f}$ = film or surface resistance.

$\frac{1}{a}$ = air-space resistance.

Fundamental Formulae

The formula of the over-all coefficient for a simple wall x inches thick is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{x}{k}} \tag{2}$$

and for a compound wall of several materials having thicknesses in inches of x_1, x_2, x_3, etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{x_1}{k_1} + \frac{x_2}{k_2} + \frac{x_3}{k_3} + etc.} \tag{3}$$

In the case of air-space construction, an air-space coefficient for each air space must be inserted in either Equation 2 or 3. Thus for a simple wall with one air space,

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{1}{a} + \frac{x}{k}} \tag{4}$$

and for a simple wall of several air spaces having conductances of a_1, a_2, a_3, etc., the coefficient is:

$$U = \frac{1}{\frac{1}{f_i} + \frac{1}{f_o} + \frac{x}{k} + \frac{1}{a_1} + \frac{1}{a_2} + \frac{1}{a_3} + etc.} \tag{5}$$

With certain special forms of materials which have irregular air spaces (such as hollow tile) or are otherwise non-homogeneous, it is necessary to use the conductance (C) for the unit construction, in which case $\frac{x}{k}$ is replaced by $\frac{1}{C}$.

As in the case of the simple wall, f_i and f_o are always the inside and outside surface coefficients for the two materials in contact with air. If the air is still (no wind), then for the same material f_i and f_o are the same, and $f_i = f_o$; but, if the outside air is in motion, then f_o is always greater than f_i and will increase as the wind velocity increases. Values for f_i in still and moving air have been determined for various building materials at the University of Minnesota under a coöperative research agreement with the Society[1]. The range of values for ordinary building materials is comparatively small and for practical purposes may be assumed constant for either still air or any given wind velocity, particularly in view of the fact that the surface resistances usually comprise only a small part of the total resistance of the construction, except in the case of thin, highly conductive walls.

TABLE 1. CONDUCTANCES OF AIR SPACES[a] AT VARIOUS MEAN TEMPERATURES

Mean Temp Deg Fahr	Conductances of Air Spaces for Various Widths in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.547	1.433	1.340	1.320	1.310
100	2.990	1.813	1.600	1.486	1.380	1.362	1.350
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
120	3.167	1.928	1.700	1.580	1.467	1.445	1.435
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
140	3.340	2.035	1.800	1.680	1.550	1.530	1.519
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

[a]Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

The conductances of air spaces at various mean temperatures and widths, for ordinary building materials, are given in Table 1. These results were likewise obtained at the University of Minnesota under a coöperative research agreement with the Society.

Values for k and C, the conductivity and conductance of building materials and insulations, are given in Tables 2, 3, 4, 5 and 6, and are taken from the published values of various investigators. It should be noted that values of k and C as well as U are dependent on the temperature range, and it is therefore desirable that the investigator determine heat-transmission values under conditions approximating those existing under actual conditions. Recommended values for calculating the coefficients of transmission of various types of construction are given in Table 7.

[1]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930). See also references at end of chapter.

While most building materials have surfaces which show similar characteristics as far as the transmission of heat is concerned, it is a well-known fact that certain surfaces such as aluminum bronze, gold bronze, aluminum foil, or in fact any metallic, highly polished surface presents a greater resistance to heat transmission than the surface of the average building material.

The greater heat resistance of such metallic surfaces is due primarily to their higher reflectivity and consequent lower emissivity of radiant heat. The use of multiple layers of metallic surfaces, combined with air spaces of low resistance, provides a definite insulating effect. Factors for single air spaces of various thicknesses bounded by aluminum foil are given in the accompanying tabulation:

CONDUCTANCES OF AIR SPACES BOUNDED BY ALUMINUM FOIL UNDER VARIOUS CONDITIONS AND FOR VARIOUS WIDTHS

CHARACTER OF SURFACES	WIDTH OF AIR SPACE—INCHES														AUTHORITY
	0.25	0.28	0.33	0.35	0.375	0.4	0.5	0.62	0.675	0.7	0.75	1.00	1.50	3⅝	
Plain Foil To Plain Foil			0.69[b] 0.64[c] 0.65[d] 0.59[e]				0.62[a] 0.57[b] 0.54[c] 0.54[d] 0.43[e]	0.55[b]			0.53[a] 0.55[b] 0.46[c] 0.46[d]	0.48[a] 0.56[b] 0.41[c] 0.42[d]	0.42[e]		E. R. Queer,* Pennsylvania State College
	0.82[v] 0.88[w] 0 94[x]	0.74[s] 0.81[t] 0.83[u]		0.61[p] 0.65[q] 0.70[v]		0.55[l] 0.60[m] 0.63[n]	0.45[i] 0.47[j] 0.52[k]			0.37[x] 0.44[g] 0.47[h]					Ralph B. Mason†
	0.88[z] 1.00[ee]		0.75[ff]		0.61[z]		0.50[y] 0.48[aa] 0.52[gg] 0.52[hh]		0.47[bb] 0.49[gg]		0.48[cc] 0.49[ff]	0.48[dd] 0.46[ff]			J. L. Gregg**
					0.64[ii] 0.72[ii]										Prof. G. B. Wilkes, Mass. Inst. of Tech.
Crumbled Foil Surfaces	1.40		1.11				0.74					0.41			J. L. Gregg**
					0.80 0.88										Prof. G. B. Wilkes, Mass. Inst. of Tech.

NOTES

a—Mean Temp. 95 F; 9 in. horizontal air space.
b— " " 95 F; 9 in. vertical air space.
c— " " 65 F; " " " "
d— " " 65 F; 9 in. horizontal air space.
e— " " 65 F; 11⅝ in. vertical air space.
f— " " 97.2 F; high side, 123.5 F; low side, 70.9 F.
g— " " 129.7 F; " ", 174.2 F; " ", 70.7 F.
h— " " 152 F; " ", 232.1 F; " ", 72.0 F.
i— " " 94.7 F; " ", 122.3 F; " ", 67.1 F.
j— " " 129.7 F; " ", 191.8 F; " ", 67.6 F.
k— " " 190.4 F; " ", 258.8 F; " ", 68.4 F.
l— " " 92.5 F; " ", 121.7 F; " ", 63.3 F.
m— " " 127.1 F; " ", 190.8 F; " ", 63.5 F.
n— " " 162.9 F; " ", 262.3 F; " ", 63.6 F.
p— " " 106.0 F; " ", 136.4 F; " ", 75.5 F.
q— " " 145.0 F; " ", 214.2 F; " ", 75.7 F.
r— " " 182.6 F; " ", 290.0 F; " ", 75.3 F.
s— " " 105.6 F; " ", 135.4 F; " ", 75.8 F.
t— " " 141.5 F; " ", 207.0 F; " ", 76.0 F.
u— " " 177.0 F; " ", 278.3 F; " ", 75.8 F.
v— " " 99.0 F; " ", 129.7 F; " ", 68.2 F.
w— " " 137.6 F; " ", 205.8 F; " ", 69.4 F.
x— " " 174.7 F; " ", 279.6 F; " ", 69.8 F.
y— " " 105 F; wooden separators occupying 6% of space
z— " " 107 F; " " " " " "
aa— " " 106 F; " " " " " "
bb— " " 100 F; " " " " " "
cc— " " 99 F; " " " " " "
dd— " " 92 F; " " " " " "
ee— " " 93 F; " " " 8% " "
ff— " " 92 F; " " " " " "
gg— " " 98 F; " " " " " "
hh— " " 97 F; fiber insulation separators occupying 8% of space.
ii— " " 115 F;

*From data in paper entitled, Importance of Radiation in Heat Transfer Through Air Spaces (*Heating, Piping and Air Conditioning*, November, 1931).
†Computed from data in article entitled, Thermal Insulation with Aluminum Foil (*Industrial and Engineering Chemistry*, March, 1933).
**Computed from data in paper entitled, Properties of Metal Foil as an Insulation (*Refrigerating Engineering*, May, 1932).

Coefficients of transmission of various types of wall, ceiling, floor and roof construction with aluminum insulation can be readily calculated. The present installation practice indicates that air spaces of ½ in. to 1½ in. are preferred but manufacturers' recommendations should be closely followed in the application of aluminum foil insulation.

TABLE 2. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATIONS

Tests Conducted at the U. S. Bureau of Standards.[p] Insulation Tests Based on Samples Submitted by Manufacturers

Note.—THE COEFFICIENTS IN TABLES 2–7, INCLUSIVE, ARE EXPRESSED IN BTU PER HOUR PER SQUARE FOOT, PER 1 DEG F, PER 1 IN. THICKNESS UNLESS OTHERWISE INDICATED.

MATERIAL	DESCRIPTION	DENSITY (LB PER CU FT)	MEAN TEMP (DEG FAHR)	CONDUCTIVITY[a] (k) OR CONDUCTANCE (C)
Asbestos Wood	Asbestos and cement compressed	123.0	86	2.70
Asbestos Mill Board	Pressed asbestos	60.5	86	0.84
Balsam Wool[h]	Chemically treated wood fiber between layers of paper	2.2	90	0.27
Cabots Quilt[h]	Eel grass between Kraft paper	4.6	90	0.26
Cabots Quilt[h]	Eel grass between Kraft paper	3.4	90	0.25
Celotex	Rigid insulation made from sugar cane fiber	13.2	90	0.34
Corkboard	Pure; no added binder	14.0	90	0.34
Corkboard	Pure; no added binder	10.6	90	0.30
Corkboard	Pure; no added binder	7.0	90	0.27
Corkboard (Eureka)	Asphaltic binder	14.5	90	0.32
Dry Zero[h]	Kapok between burlap or paper	1.0	90	0.24
Eagle Insulating Wool		9.4	103	0.27
Fibrofelt[h]	Flax and rye fiber	13.6	90	0.32
Flaxlinum[h]	Flax fiber	13.0	90	0.31
Gyplap	Gypsum between layers of heavy paper (½ in. thick)	53.5	90	2.60[b]
Hairinsul[h]	75% hair; 25% jute	6.3	90	0.27
Hairinsul[h]	50% hair; 50% jute	6.1	90	0.26
Hair Felt[h]	Felted cattle hair	13.0	90	0.26
Hair Felt[h]	Felted cattle hair	11.0	90	0.26
Insulex or Pyrocell	Cellular gypsum—dry	30.0	90	1.00
Insulex or Pyrocell	Cellular gypsum—dry	24.0	90	0.77
Insulex or Pyrocell	Cellular gypsum—dry	18.0	90	0.59
Insulex or Pyrocell	Cellular gypsum—dry	12.0	90	0.44
Insulite	Rigid insulation made from wood pulp	16.9	90	0.34
Linofelt[h]	Flax fibers between paper	4.9	90	0.28
Lith	Rock wool, flax and straw pulp with binder	14.3	90	0.40
Magnesia (Rigid)	85% magnesia, 15% asbestos	19.3	86	0.51
Plaster	Gypsum	46.2	86	2.32[i]
Regranulated cork	About 3/16 in. particles	8.1	90	0.31
Rock cork	Rock wool block with binders	14.5	77	0.33
Rock wool	Fibrous material, made from rock	10.0	90	0.27
Rock wool	Fibrous material, made from rock	14.0	90	0.28
Rock wool	Fibrous material, made from rock	18.0	90	0.29
Rock wool	Fibrous material, made from rock	21.0	90	0.30
Sawdust	Ordinary	....	86	1.04
Shavings	Ordinary	....	86	0.71
Sheetrock	Gypsum mixed with sawdust between layers of heavy paper (0.39 in. thick)	60.7	90	3.60[b]
Thermofelt[h]	Jute and asbestos fibers, felted	10.0	90	0.37
Thermofelt[h]	Hair and asbestos fibers, felted	7.8	90	0.28
Thermofill[h]	Dry, fluffy, flaked gypsum	34.0	90	0.60
Thermofill[h]	Dry, fluffy, flaked gypsum	26.0	90	0.52
Thermofill[h]	Dry, fluffy, flaked gypsum	19.8	90	0.35
Torfoleum	Peat moss compressed into sheet form	10.2	91.5	0.29
Woods:				
Balsa wood	Across grain	20.0	90	0.58
Balsa wood	Across grain	8.8	90	0.38
Balsa wood	Across grain	7.3	90	0.33
Cypress	Across grain	28.7	86	0.67
Maple	Across grain	44.3	86	1.10
Mahogany	Across grain	34.3	86	0.90
Virginia pine	Across grain	34.3	86	0.96
White pine	Across grain	31.2	86	0.78

[a]In addition to the conductivity values for the authorities listed, considerable work of importance pertaining to the heat transmission of various types of construction and materials has been done by the late Prof. John R. Allen and the late Prof. A. J. Wood of the Engineering Experiment Station of Pennsylvania State College.

[b]For thickness stated or used in construction, not per 1 in. thickness.

[h]Not compressed.

[i]The conductivity of plaster varies with the composition. Note range of values from 2.32 to 8.0. On account of the comparatively high conductivity of plaster and the fact that it is seldom applied more than ¾ in. thick, this material does not appreciably affect the over-all transmission of a construction, excepting in the case of thin uninsulated walls.

[p]See Chapter LX, by Chas. H. Herter of the Report of the Insulation Committee, *A.S.R.E.*, Annual Meeting 1922, Revised to 1924, entitled Heat Transmission of Insulating Materials for a more comprehensive collection of heat transmission data relating to building and insulating materials.

The majority of the conductivities and conductances of the building materials and insulations given in Tables 2, 3, 4, 5 and 6 were determined by the hot-plate method of testing[2]. Attention is called to the fact that conductivities per inch of thickness of materials or insulations do not afford a true basis for comparison, although they are frequently used for that purpose. Correct comparisons should take into consideration many different factors, including conductivities or conductances, thicknesses installed and manner of installation, while the selection of an insulation should also give consideration to structural qualities, as well as material and application costs. Fire, vermin, and rot resistance are other important factors to be considered when comparing materials. At present there is no universally recognized method of rating insulations. Conductivities and conductances of building materials and insulations are useful to the heating engineer in determining over-all coefficients of heat transmission of walls, floors, roofs and ceilings.

TABLE 3. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATIONS

Based on Tests Conducted at the University of Illinois, By A. C. Willard, L. C. Lichty and L. A. Harding [p]

MATERIAL	DESCRIPTION	DENSITY (LB PER CU FT)	MEAN TEMP (DEG FAHR)	CONDUCTIVITY (k) OR CONDUCTANCE (C) [a]
Asbestos	Sheet	48.3	110	0.29
Asbestos board	Corrugated	20.4	110	0.48
Brickwork	Mortar bond and dry conditions	132.0	100	4.00
Brickwork	Damp or wet	......	.....	5.00[j]
Cement mortar		......	.....	8.00[f]
Concrete	Stone 1-2-4 mix	140.0	110	8.30
Corkboard	Pure; no added binder	9.7	.....	0.32
2 in. hollow clay tile, ½ in. plaster both sides		120.0	110	1.00[b]
4 in. hollow clay tile, ½ in. plaster both sides		127.0	100	0.60[b]
6 in. hollow clay tile, ½ in. plaster both sides		124.3	105	0.47[b]
Magnesia board		13.5	110	0.51
Plaster	Cement	......	.....	8.00[i]
Roofing[h]	Built-up bitumen and felt, gravel or slag surfaced	......	.....	1.325[b]
Roofing[h]	Built-up bitumen and felt, gravel or slag surfaced	......	.....	5.30[d-b]
Stucco		......	.....	8.00[f]
Wood (Fir, one-surface finished)	Across grain	33.4	.....	1.00

[a]In addition to the conductivity values for the authorities listed, considerable work of importance pertaining to the heat transmission of various types of construction and materials has been done by the late Prof. John R. Allen and the late Prof. A. J. Wood of the Engineering Experiment Station of Pennsylvania State College.

[b]For thickness stated or used in construction, not per 1 in. thickness.

[d]Calculated from 2 in. tile tests.

[f]Cement mortar and stucco assumed same as cement plaster.

[h]Roofing, 0.15 in. thick (1.34 lb per sq ft), covered with gravel (0.83 lb per sq ft), combined thickness assumed 0.25.

[i]The conductivity of plaster varies with the composition. Note range of values from 2.32 to 8.0. On account of the comparatively high conductivity of plaster and the fact that it is seldom applied more than ¾ in. thick, this material does not appreciably affect the over-all transmission of a construction, excepting in the case of thin uninsulated walls.

[j]Recommended value. See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.

[p]See Chapter LX, by Chas. H. Herter of the Report of the Insulation Committee, *A.S.R.E.*, Annual Meeting, 1922, Revised to 1924, entitled, Heat Transmission of Insulating Materials for a more comprehensive collection of heat transmission data relating to building and insulating materials.

[2]See Standard Test Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928). See also Chapter 40.

TABLE 4. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATIONS

Tests Conducted at Armour Institute of Technology, by J. C. Peebles.[p] *Insulation Tests Based on Samples Submitted by Manufacturers*

MATERIAL	DESCRIPTION	DENSITY (LB. PER CU. FT.)	MEAN TEMP. (DEG. FAHR.)	CONDUCTIVITY (k) OR CONDUCTANCE (C)[a]
Aerocrete	Cellular concrete	40.0	75	1.06
Aerocrete	Cellular concrete	50.0	75	1.44
Aerocrete	Cellular concrete	60.0	75	1.80
Aerocrete	Cellular concrete	70.0	75	2.18
Arborite	Rigid insulation made from wood pulp	15.2	....	0.33
Asbestos shingles		65.0	75	6.00[b]
Asphalt shingles		70.0	75	6.50[b]
Balsam Wool[h]	Chemically treated wood fibre between plain paper	3.62	70	0.25
Beaver Insulating Board	Cane fibre commercial thickness ½ in.	13.8	75	0.33
Calicel	Granular fill insulation made from combined silicate of lime and alumina	4.2	72	0.24
Celotex	Rigid insulation made from sugar cane fibre	13.5	70	0.33
Cincrete Block, 8 in. Hollow	Treated cinder aggregate concrete	....	....	0.33[b]
Concrete	Stone	145.0	75	6.30
Concrete	Cinder	110.0	75	5.20
Donnacona Board	Rigid insulation made from wood fiber	15.9	72	0.33
Dry Zero Blanket[h]	Pliable slab form of insulation made from ceiba fibres	1.9	75	0.23
Dry Zero Blanket[h]	Pliable slab form of insulation made from ceiba fibres	1.6	75	0.24
Dry Zero Blanket[h]	Pliable slab form of insulation made from ceiba fibres	1.5	75	0.24
Flax-li-num[h]	Flax fibre	12.1	70	0.30
Glass Wool	Fibrous material 25 to 30 microns in diameter made from virgin bottle glass	1.5	75	0.27
Glass Wool	Ditto, 2 to 3 microns in diameter	0.85	75	0.25
Haydite—Concrete	Heat-treated clay aggregate standard blocks 8x8x16 in.	73.0	....	1.62
Homasote	Made from wood and other vegetable fibres chemically treated	25.0	75	0.375
Inso Board	Rigid insulation made from wheat straw	17.0	68	0.33
Insulating Plaster	Insulating plaster, 9/10 in. thick, applied to ⅜ in. plaster board base	54.0	75	1.07[b]
Insulite	Rigid insulation made from wood pulp	16.5	70	0.34
Keystone Hair[h]	Hair felt between layers of paper; ¼ in. thick	11.0	75	0.25
Lith	Rock wool, flax and straw pulp with binder	14.5	75	0.38
Linoboard	Slab form of insulation made from rock wool and vegetable fibres	9.9	72	0.288
		11.5	72	0.311
Maftex	Rigid insulation made from licorice roots	16.1	81	0.34
Maizewood	Rigid insulation made from cornstalks	15.0	70.5	0.323
Maple Flooring	Across grain	40.0	75	1.20
Masonite	Rigid insulation made from exploded wood fibre	19.8	75	0.33
Plaster Board	Gypsum between layers of heavy paper	62.8	70	1.41
Pyrocell or Insulex	Cellular gypsum—dry	30.0	75	0.92
Pyrocell or Insulex	Cellular gypsum—dry	12.0	75	0.40
Red Top Insulating Wool	Fibrous material made from dolomite and silica	1.5	75	0.27
Roofing	Composition or prepared	......	75	6.50[b]
Temlok	Rigid insulation made from wood fiber	15.0	70	0.33
Therm-A-Pad[h]	Flexible insulation made from jute	6.7	75	0.25
Thermatex	Rigid insulation made from wood fiber	8.5	72	0.294
Thermax	Made from shredded wood and cement	24.2	72	0.46
Thermofil	Dry, fluffy, flaked gypsum	24.0	75	0.48
Thermofil	Dry, fluffy, flaked gypsum	18.0	75	0.34
Thermasote "A" Insulating Board	Made from wood fibre, chemically treated	20.0	70	0.353
Torfoleum	Peat moss compressed into sheet form	11.0	70	0.26
Weatherwood	Rigid insulation made from hard wood fibres	15.2	70	0.32
Wood Lath and Plaster[k]	Lime plaster	....	75	2.0[b]
Yellow Pine	Across grain	.....	.	1.00

[a]In addition to the conductivity values for the authorities listed, considerable work of importance pertaining to the heat transmission of various types of construction and materials has been done by the late Prof. John R. Allen and the late Pr. A. J. Wood of the Engineering Experiment Station of Pennsylvania State College.

[b]For thickness stated or used in construction, not per 1 in. thickness. [h]Not compressed.

[k]Thickness of lime plaster and wood lath from back of lath to face of plaster, about ⅞ in.

[p]See Chapter LX, by Chas. H. Herter of the Report of the Insulation Committee, *A.S.R.E.*, Annual Meeting, 1922 Revised to 1924, entitled, Heat Transmission of Insulating Materials for a more comprehensive collection of heat transmission data relating to building and insulating materials.

TABLE 5. CONDUCTIVITIES (k) AND CONDUCTANCES (C) OF BUILDING MATERIALS AND INSULATIONS

Based on Tests Conducted at the University of Minnesota, By F. B. Rowley [p]

MATERIAL	DESCRIPTION	DENSITY (LB PER CU FT)	MEAN TEMP (DEG FAHR)	CONDUCTIVITY (k) OR CONDUCTANCE (C) [a]
Concrete	Stone 1-2-4 mix	143.0	68.8	9.46
Dry Zero	Pliable slab form of insulation made from ceiba fibres	....	....	0.23
Fir sheathing and building paper	....	....	30.0	0.71[b]
Fir sheathing, building paper and pine lap siding	....	....	20.0	0.50[b]
Fir sheathing, building paper and stucco	....	....	20.0	0.82[b]
Gypsum Tile	Solid	51.8	69.9	1.66
Gypsum Tile	Solid	75.6	75.9	2.96
Gypsum Fibre Concrete	87½% gypsum and 12½% wood chips	51.2	74.4	1.66
Lath and ⅝ in. plaster	Total thickness ¾ in.	....	70.0	2.50[b]
Masonite	Rigid insulation made from exploded wood fibre	17.9	77.6	0.32
Pine lap siding and building paper	Lap siding 4 in. wide	....	15.5	0.85[b]
Plaster	Thickness ⅝ in.	....	73.0	8.8[b]
Sheet Rock Pyrofill Roofing, 2½ in. thick	Plaster board, gypsum fibre concrete and 3-ply roof covering	52.4	76.0	0.58[b]

[a]In addition to the conductivity values for the authorities listed, considerable work of importance pertaining to the heat transmission of various types of construction and materials has been done by the late Prof. John R. Allen and the late Prof. A. J. Wood of the Engineering Experiment Station of Pennsylvania State College.

[b]For thickness stated or used in construction, not per 1 in. thickness.

[p]See Chapter LX, by Chas. H. Herter of the Report of the Insulation Committee, *A.S.R.E.*, Annual Meeting, 1922, Revised to 1924, entitled Heat Transmission of Insulating Materials for a more comprehensive collection of heat transmission data relating to building and insulating materials.

The coefficients in these tables were determined by calculations similar to those shown in Example 1, using Equations 2, 3, 4 and 5 and the values of k (or C), f_i, f_o and a indicated in Table 7. In computing heat transmission coefficients of floors laid directly on the ground (Table 15), only one surface coefficient (f_i) is used. For example, the value of U for a 1-in.

TABLE 6. MISCELLANEOUS CONDUCTIVITIES (k) OF MATERIALS[d]

Expressed in Btu per hour per square foot per degree Fahrenheit per inch thickness

MATERIAL	DESCRIPTION	MEAN TEMP. (DEG FAHR)	CONDUCTIVITY (k)t	AUTHORITY
Alfol	Aluminum foil ⅜ in. spacing	115	0.24 to 0.27	G. B. Wilkes, Mass. Inst. of Technology
Concrete	Stone 1-2-5 mix	95[b]	6.27	C. L. Norton, Boston, Mass.
Concrete	Cinder 1-2-4 mix	122	2.35	C. L. Norton, Boston, Mass.
Concrete	Various ages and mixes	....	11.35 to 16.36	A.S.H.V.E.[c] Research Laboratory
Slate	....	201	10.37	Lees and Chorlton
Ten Test	Rigid insulation made from wood fiber	52	0.33	E. A. Allcut, University of Toronto

[a]In addition to the conductivity values for the authorities listed, considerable work of importance pertaining to the heat transmission of various types of construction and materials has been done by the late Prof. John R. Allen and the late Prof. A. J. Wood of the Engineering Experiment Station of Pennsylvania State College.

[b]Hot side of plate.

[c]See A.S.H.V.E. Research paper entitled Conductivity of Concrete, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

[d]See Chapter LX, by Chas. H. Herter of the Report of the Insulation Committee, *A.S.R.E.*, Annual Meeting, 1922, Revised to 1924, entitled, Heat Transmission of Insulating Materials for a more comprehensive collection of heat transmission data relating to building and insulating materials.

yellow pine floor (actual thickness, 25/32 in.) placed directly on 6 in. concrete on the ground, is determined as follows:

$$U = \frac{1}{\frac{1}{1.65} + \frac{0.781}{0.80} + \frac{6.0}{12.0}} = 0.48$$ Btu per hour per square foot per degree difference in temperature between the ground and the air immediately above the floor.

The thicknesses upon which the coefficients in Tables 8 to 18, inclusive, are based are as follows:

Brick veneer	4 in.
Plaster and metal lath	3/4 in.
Plaster (on wood lath, plasterboard, rigid insulation, board form, or corkboard)	1/2 in.
Slate (roofing)	1/2 in.
Stucco on wire mesh reinforcing	1 in.
Tar and gravel or slag-surfaced built-up roofing	3/8 in.
1-in. Lumber (S-2-S)	25/32 in.
1½-in. Lumber (S-2-S)	1 5/16 in.
2-in. Lumber (S-2-S)	1 5/8 in.
2½-in. Lumber (S-2-S)	2 1/8 in.
3-in. Lumber (S-2-S)	2 5/8 in.
4-in. Lumber (S-2-S)	3 5/8 in.
Finish flooring (maple or oak)	13/16 in.

Solid brick walls are based on 4-in. face brick and the remainder common brick. Stucco is assumed to be 1 in. thick on masonry walls. Where metal lath and plaster is specified, the metal lath is neglected.

Rigid insulation refers to the so-called board form which may be used structurally, such as for sheathing. Flexible insulation refers to the blankets, quilts or semi-rigid types of insulation.

Actual thicknesses of lumber are used in the computations rather than nominal thicknesses. The computations for wood shingle roofs applied over wood stripping are based on 1 by 4 in. wood strips, spaced 2 in. apart. Since no reliable figures are available concerning the conductivity of Spanish and French clay roofing tile, of which there are many varieties, the figures for such types of roofs were taken the same as for slate roofs, as it is probable that the values of U for these two types of roofs will compare favorably.

The coefficients of transmission of the pitched roofs in Table 17 apply where the roof is over a heated attic or top floor, such that the heat passes directly through the roof structure including whatever finish, if any, is applied to the underside of the roof rafters.

Combined Coefficients of Transmission

If the attic is unheated, the roof structure and ceiling of the top floor must both be taken into consideration, and the combined coefficient of transmission determined. The formula for calculating the combined coefficient of transmission of a top-floor ceiling, unheated attic space and pitched roof, per square foot of roof area, is as follows:

$$U = \frac{U_r \times U_{ce}}{n \times U_r + U_{ce}} \tag{8}$$

where

U_r = coefficient of transmission of the roof.
U_{ce} = coefficient of transmission of the ceiling.
n = the ratio of the area of the roof to the area of the ceiling.

Example 1. Calculate the coefficient of transmission (U) of an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, based on an outside wind exposure of 15 mph. It is assumed that the outside course is of face brick having a conductivity of 9.20, and that the inside course is of common brick having a conductivity of 5.0, the thicknesses each being 4 in. The conductivity of the plaster is assumed to be 3.3, and the inside and outside surface coefficients are assumed to average 1.65 and 6.00, respectively, for still air and a 15 mph wind velocity.

Solution. k (face brick) = 9.20; x = 4.0 in.; k (common brick) = 5.0; x = 4.0 in.; k (plaster) = 3.3; x = ½ in.; f_i = 1.65; f_o = 6.0. Therefore,

$$U = \frac{1}{\frac{1}{6.0} + \frac{4.0}{9.20} + \frac{4.0}{5.0} + \frac{0.5}{3.3} + \frac{1}{1.65}}$$

$$= \frac{1}{0.167 + 0.435 + 0.80 + 0.152 + 0.606}$$

= 0.46 Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides.

Computed Transmission Coefficients

Computed heat transmission coefficients of many common types of building construction are given in Tables 8 to 18, inclusive, each construction being identified by a serial number. For example, the coefficient of transmission (U) of an 8-in. brick wall and ½ in. of plaster is 0.46, and the number assigned to a wall of this construction is *1-B*, Table 8.

In using this formula, a correction factor must be applied. As the amount of heat transferred through an air space is proportional to the difference of the fourth powers of the absolute temperatures of the surfaces enclosing the air space, a greater amount of heat is absorbed or emitted by radiation by the surfaces enclosing an unheated attic than by the surfaces of a wall or ceiling in a room under still-air conditions, where the surrounding objects are only slightly higher in temperature than the interior surfaces of the walls and ceiling. For example, the average coefficient of a surface in still air is 1.65 Btu per hour per square foot per degree Fahrenheit, whereas the average coefficient of an air space in an outside wall is about 1.10 Btu per hour per square foot per degree Fahrenheit difference between the two surfaces, at a mean temperature of 40 F. An air space coefficient of 1.10 is equivalent to a surface coefficient of 2.20 for each of the two surfaces enclosing the air space, where the over-all transmission is computed by using the coefficients of the two surfaces enclosing the air space instead of the coefficient of the air space itself. Hence, in determining the values of U_r and U_{ce} to be used in the formula, the coefficients for the surfaces of the roof and ceiling enclosing the attic should be increased to allow for the additional amount of heat transferred by radiation, and a coefficient of 2.20 may be used with sufficient accuracy for each of these surfaces, although in very precise work a correction should be made to allow for the fact that the area of a pitched roof over an unheated attic is greater than the area of the ceiling,

TABLE 7. RECOMMENDED CONDUCTIVITIES AND CONDUCTANCES FOR COMPUTING HEAT TRANSMISSION COEFFICIENTS

Conductivities are expressed in Btu per hour per square foot per degree Fahrenheit per inch thickness. Conductances are indicated by asterisks (*) and are for the thickness or condition stated, not per inch.

MATERIAL	CONDUCTIVITY OR CONDUCTANCE	MATERIAL	CONDUCTIVITY OR CONDUCTANCE
Brick, Common	5.00	Plaster Board (½ in.)	2.82*
Brick, Face	9.2	Roofing	
Cement Mortar	12.00	Asbestos Shingles	6.00*
Cinder Concrete	5.20	Asphalt or Composition Roofing	6.50*
Cinder Blocks[a] (8 in.)	0.62*	Built-up, ⅜ in. thick	3.53*
Cinder Blocks[a] (12 in.)	0.51*	Slate Shingles	10.37
Concrete Blocks[a] (8 in.)	1.0*	Wood Shingles (see woods)	
Concrete Blocks[a] (12 in.)	0.80*	Stone	12.50
Concrete	12.00	Stucco	12.00
Gypsum Fiber Concrete	1.66	Tile or Terrazzo	12.00
Hollow Clay Tile (4 in.)	1.00*	Wood Lath and Plaster	2.50*
Hollow Clay Tile (6 in.)[b]	0.64*	Woods	
Hollow Clay Tile (8 in.)[b]	0.60*	1-in. Fir sheathing, building paper and yellow pine lap siding	0.50*
Hollow Clay Tile (10 in.)[b]	0.58*		
Hollow Clay Tile (12 in.)[b]	0.40*		
Hollow Clay Tile (16 in.)[b]	0.31*	1-in. Fir sheathing and building paper	0.82*
Hollow Gypsum Tile (4 in.)	0.46*		
Insulations			
Corkboard	0.30	Yellow pine lap siding	1.28*
Flexible	0.27	Yellow pine or fir	0.80
Flaked Gypsum (24 lb)	0.48	Maple or oak	1.15
Rigid Insulation	0.33	Shingles, wood	1.28*
Rock Wool	0.30	Air spaces	1.10*
Plaster (Gypsum)	3.3	Surfaces, still air (f_i)	1.65*
Plaster Board (⅜ in.)	3.73*	Surfaces, 15 mph (f_o)	6.00*

*For thickness or condition stated, not per 1 in.

[a]One air cell in the direction of heat flow.

[b]The 6-in., 8-in. and 10-in hollow tile figures are based on two cells in the direction of heat flow. The 12-in. hollow tile is based on three cells in the direction of heat flow. The 16-in. hollow tile consists of one 10-in. and one 6-in. tile, each having two cells in the direction of heat flow.

and hence, the amount of heat absorbed by radiation by each square foot of roof surface is less than is given off by radiation by each square foot of ceiling surface.

Example 2. Determine the combined coefficient of transmission of a roof constructed of asbestos shingles applied over wood sheathing on rafters, an unheated attic, and a wood lath and plaster ceiling, based on a roof having a one-third pitch, for which the value of n is 1.2.

$$U_r = \frac{1}{\frac{1}{6.0} + \frac{1}{2.20} + \frac{1}{6.0} + \frac{0.781}{0.80}} = 0.567$$

$$U_{ce} = \frac{1}{\frac{1}{1.65} + \frac{1}{2.20} + \frac{1}{2.50}} = 0.685$$

Substituting these values in Equation 6:

$$U = \frac{0.567 \times 0.685}{1.2 \times 0.567 + 0.685} = 0.28$$

Btu per hour *per square foot of roof area per* degree difference in temperature between the air near the under side of the ceiling and the outside air.

If the unheated attic space between the roof and ceiling has no dormers, windows or vertical wall surfaces, the combined coefficients may be used for determining the heat loss through the roof construction between the attic and top-floor ceiling, but it should be noted that these coefficients

TABLE 8. COEFFICIENTS OF TRANSMISSION (U) OF MASONRY WALLS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph

INTERIOR FINISH

UNINSULATED WALLS

COLUMN A. Plain walls—no interior finish.
COLUMN B. Plaster (½ in.) on walls.
COLUMN C. Plaster on wood lath—furred.
COLUMN D. Plaster (¾ in.) on metal lath—furred.
COLUMN E. Plaster (½ in.) on plaster board (⅜ in.)—furred.

INSULATED WALLS

COLUMN F. Plaster (½ in.) on rigid insulation (½ in.)—furred.
COLUMN G. Plaster (½ in.) on rigid insulation (1 in.)—furred.
COLUMN H. Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.).
COLUMN I. Plaster on corkboard (2 in.) set in cement mortar (½ in.).
COLUMN J. Plaster on wood lath attached to furring strips (2 in.[b])—flaked gypsum fill (1⅝ in.[b]).[g]
COLUMN K. Plaster on wood lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b]).[g]
COLUMN L. Plaster on wood lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space).

WALL No.	THICKNESS OF MASONRY (INCHES)	TYPE OF WALL	A	B	C	D	E	F	G	H	I	J	K	L
1	8	Solid Brick[f]	0.50	0.46	0.30	0.32	0.30	0.22	0.16	0.14	0.11	0.17	0.13	0.20
2	12		0.36	0.34	0.24	0.25	0.24	0.19	0.14	0.12	0.10	0.15	0.12	0.17
3	16		0.28	0.27	0.20	0.21	0.20	0.16	0.13	0.11	0.10	0.14	0.11	0.15
4	8	Hollow Tile[c] Stucco Exterior Finish	0.40	0.37	0.26	0.27	0.26	0.20	0.15	0.13	0.11	0.16	0.12	0.18
5	10		0.39	0.37	0.26	0.27	0.26	0.19	0.15	0.13	0.11	0.16	0.12	0.17
6	12		0.30	0.29	0.22	0.22	0.22	0.17	0.13	0.12	0.10	0.14	0.11	0.15
7	16		0.25	0.24	0.19	0.19	0.19	0.15	0.12	0.11	0.09	0.13	0.10	0.14
8	8	Limestone or Sandstone	0.71	0.64	0.37	0.39	0.37	0.25	0.18	0.15	0.12	0.20	0.14	0.22
9	12		0.58	0.53	0.33	0.34	0.33	0.23	0.17	0.14	0.12	0.18	0.13	0.21
10	16		0.49	0.45	0.30	0.31	0.30	0.22	0.16	0.14	0.11	0.17	0.13	0.19
11	24		0.37	0.35	0.25	0.26	0.25	0.19	0.15	0.13	0.11	0.15	0.12	0.17
12	6	Concrete[d]	0.79	0.70	0.39	0.42	0.39	0.26	0.19	0.16	0.12	0.20	0.14	0.23
13	10		0.62	0.57	0.34	0.37	0.34	0.24	0.18	0.15	0.12	0.19	0.13	0.21
14	16		0.48	0.44	0.29	0.31	0.29	0.21	0.16	0.14	0.11	0.17	0.13	0.19
15	20		0.41	0.39	0.27	0.28	0.27	0.20	0.15	0.13	0.11	0.16	0.12	0.18
16	8	Hollow Cinder Blocks[e]	0.42	0.39	0.27	0.28	0.27	0.20	0.16	0.13	0.11	0.16	0.12	0.18
17	12		0.37	0.35	0.25	0.26	0.25	0.19	0.15	0.13	0.11	0.15	0.12	0.17
18	8	Hollow Concrete Blocks[e]	0.56	0.52	0.32	0.34	0.32	0.23	0.17	0.14	0.12	0.18	0.13	0.20
19	12		0.49	0.46	0.30	0.32	0.30	0.22	0.16	0.14	0.11	0.17	0.13	0.19

[a]Computed from factors given in Table 7.
[b]Based on the actual thickness of 2 in. furring strips.
[c]The 8-in. and 10-in. tile figures are based on two cells in the direction of flow of heat. The 12-in. tile is based on three cells in the direction of flow of heat. The 16-in. tile consists of one 10-in. tile and one 6-in. tile, each having two cells in the direction of heat flow.
[d]These figures may be used with sufficient accuracy for concrete walls with stucco exterior finish.
[e]Based on one air cell in direction of heat flow.
[f]Based on 4 in. face brick and the remainder common brick.
[g]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

Table 9. Coefficients of Transmission (U) of Masonry Walls with Various Types of Veneers[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

INTERIOR FINISH

Uninsulated Walls

Column A. Plain walls—no interior finish.
Column B. Plaster (½ in.) on walls.
Column C. Plaster on wood lath—furred.
Column D. Plaster (¾ in.) on metal lath—furred.
Column E. Plaster (½ in.) on plaster board (⅜ in.)—furred.

Insulated Walls

Column F. Plaster (½ in.) on rigid insulation (½ in.)—furred.
Column G. Plaster (½ in.) on rigid insulation (1 in.)—furred.
Column H. Plaster (½ in.) on corkboard (1½ in.) set in cement mortar (½ in.).
Column I. Plaster on corkboard (2 in.) set in cement mortar (½ in.).
Column J. Plaster on wood lath attached to furring strips (2 in.)[b]—flaked gypsum fill (1⅝ in.[b]).[f]
Column K. Plaster on wood lath attached to furring strips (2 in.[b])—rock wool fill (1⅝ in.[b]).[f]
Column L. Plaster on wood lath attached to furring strips (2 in.[b])—flexible insulation (½ in.) between furring strips (one air space).

Wall No.	Type of Wall: Facing		Backing	A	B	C	D	E	F	G	H	I	J	K	L
20	4 In. Brick Veneer[d]	6"	Hollow Tile[c]	0.36	0.34	0.24	0.25	0.24	0.19	0.16	0.13	0.10	0.15	0.12	0.17
21		8"		0.34	0.33	0.24	0.25	0.24	0.18	0.14	0.12	0.10	0.15	0.11	0.17
22		10"		0.34	0.32	0.23	0.24	0.23	0.18	0.14	0.12	0.10	0.15	0.11	0.16
23		12"		0.27	0.26	0.20	0.21	0.20	0.16	0.13	0.11	0.094	0.13	0.11	0.15
24	4 In. Brick Veneer[d]	6"	Concrete	0.57	0.53	0.33	0.35	0.33	0.23	0.17	0.14	0.12	0.18	0.13	0.20
25		10"		0.48	0.45	0.30	0.31	0.30	0.22	0.16	0.14	0.11	0.17	0.13	0.19
26		16"		0.39	0.37	0.26	0.27	0.26	0.19	0.15	0.13	0.11	0.16	0.12	0.17
27	4 In. Brick Veneer[d]	8"	Cinder Blocks[e]	0.35	0.33	0.24	0.25	0.24	0.18	0.14	0.12	0.10	0.15	0.12	0.17
28		12"		0.31	0.30	0.22	0.23	0.22	0.17	0.14	0.12	0.10	0.14	0.11	0.16
29	4 In. Brick Veneer[d]	8"	Concrete Blocks[e]	0.44	0.42	0.28	0.30	0.28	0.21	0.16	0.13	0.11	0.17	0.12	0.18
30		12"		0.40	0.38	0.26	0.28	0.26	0.20	0.15	0.13	0.11	0.16	0.12	0.18
31	4 In. Cut-Stone Veneer[d]	8"	Common Brick	0.37	0.35	0.25	0.26	0.25	0.19	0.15	0.13	0.10	0.15	0.12	0.17
32		12"		0.28	0.27	0.21	0.21	0.21	0.16	0.13	0.12	0.097	0.14	0.11	0.15
33		16"		0.23	0.22	0.18	0.18	0.18	0.14	0.12	0.11	0.090	0.12	0.10	0.13
34	4 In. Cut-Stone Veneer[d]	6"	Hollow Tile[c]	0.37	0.35	0.25	0.26	0.25	0.19	0.15	0.13	0.10	0.15	0.12	0.17
35		8"		0.36	0.34	0.24	0.25	0.24	0.19	0.15	0.13	0.10	0.15	0.12	0.17
36		10"		0.35	0.33	0.24	0.25	0.24	0.18	0.14	0.12	0.10	0.15	0.12	0.17
37		12"		0.28	0.26	0.20	0.21	0.20	0.16	0.13	0.11	0.096	0.14	0.11	0.15
38	4 In. Cut-Stone Veneer[d]	6"	Concrete	0.61	0.56	0.34	0.36	0.34	0.24	0.18	0.15	0.12	0.19	0.13	0.21
39		10"		0.51	0.47	0.31	0.32	0.31	0.22	0.17	0.14	0.11	0.17	0.13	0.20
40		16"		0.41	0.38	0.26	0.25	0.26	0.20	0.15	0.13	0.11	0.16	0.12	0.18

[a]Computed from factors given in Table 7.
[b]Based on the actual thickness of 2 in. furring strips.
[c]The 6-in., 8-in. and 10-in. tile figures are based on two cells in the direction of heat flow. The 12-in. tile is based on three cells in the direction of heat flow.
[d]Calculations include cement mortar (½ in.) between veneer or facing and backing.
[e]Based on one air cell in direction of heat flow.
[f]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) OF VARIOUS TYPES OF FRAME CONSTRUCTION[a]

These coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on a wind velocity of 15 mph.

INTERIOR FINISH

NO INSULATION BETWEEN STUDDING

COLUMN A. Plaster on wood lath on studding.
COLUMN B. Plaster (3/4 in.) on metal lath on studding.
COLUMN C. Plaster (1/2 in.) on plaster board (3/8 in.) on studding.
COLUMN D. Plaster (1/2 in.) on rigid insulation (1/2 in.) on studding.
COLUMN E. Plaster (1/2 in.) on rigid insulation (1 in.) on studding.
COLUMN F. Plaster (1/2 in.) on corkboard (1 1/2 in.) on studding.
COLUMN G. Plaster (1/2 in.) on corkboard (2 in.) on studding.

INSULATION BETWEEN STUDDING

COLUMN H. Plaster on wood lath[b] on studding—flaked gypsum fill (3 5/8 in.[c]) between studding.[g]
COLUMN I. Plaster on wood lath[b] on studding—rock wool fill (3 5/8 in.[c]) between studding.[g]
COLUMN J. Plaster on wood lath[b] on studding—flexible insulation (1/2 in.) between *studding and in contact with sheathing.*

WALL No.	EXTERIOR FINISH	TYPE OF SHEATHING	A	B	C	D	E	F	G	H	I	J
41	Wood Siding or Clapboard	1 in. Wood[d]	0.25	0.26	0.25	0.19	0.15	0.11	0.095	0.093	0.066	0.17
42		1/2 in. Rigid Insulation	0.23	0.24	0.23	0.18	0.14	0.11	0.093	0.091	0.064	0.16
43		1/2 in. Plaster Board	0.31	0.33	0.31	0.22	0.17	0.13	0.10	0.10	0.070	0.20
44	Wood Shingles	1 in. Wood[d]	0.25	0.26	0.25	0.19	0.15	0.11	0.095	0.093	0.066	0.17
45		1/2 in. Rigid Insulation[e]	0.19	0.20	0.19	0.15	0.12	0.10	0.085	0.084	0.061	0.14
46		1/2 in. Plaster Board[e]	0.24	0.25	0.24	0.19	0.15	0.11	0.095	0.093	0.065	0.17
47	Stucco	1 in. Wood[d]	0.30	0.31	0.30	0.22	0.16	0.12	0.10	0.10	0.069	0.19
48		1/2 in. Rigid Insulation	0.27	0.29	0.27	0.20	0.16	0.12	0.099	0.097	0.067	0.18
49		1/2 in. Plaster Board	0.40	0.43	0.40	0.26	0.19	0.14	0.11	0.11	0.073	0.23
50	Brick[f] Veneer	1 in. Wood[d]	0.27	0.28	0.27	0.20	0.15	0.12	0.098	0.096	0.067	0.18
51		1/2 in. Rigid Insulation	0.25	0.26	0.25	0.19	0.15	0.11	0.096	0.094	0.066	0.17
52		1/2 in. Plaster Board	0.35	0.37	0.35	0.24	0.18	0.13	0.11	0.11	0.071	0.21

[a]Computed from factors given in Table 7.
[b]These coefficients may also be used with sufficient accuracy for plaster on metal lath or plaster on plaster board.
[c]Based on the actual width of 2 by 4 studding, namely 3 5/8 in.
[d]Yellow pine or fir—actual thickness about 25/32 in.
[e]Furring strips between wood shingles and sheathing.
[f]Small air space and mortar between building paper and brick veneer neglected.
[g]A waterproof membrane should be provided between the outer material and the insulation fill to prevent possible wetting by absorption and a subsequent lowering of efficiency.

should be multiplied by the roof area and not by the ceiling area. If the unheated attic contains windows, ventilators or vertical wall surfaces, which would tend to reduce temperature in the attic to a temperature approaching or equaling the outside temperature, the roof should be neglected and only the top-floor ceiling construction and the corresponding ceiling area taken into consideration, using the coefficients given in Tables 13 or 14. Tests made in Pittsburgh by Professor Humphreys indicated that for this type of attic, an attic temperature should be taken which is an average between the inside and the outside temperature.

TABLE 11. COEFFICIENTS OF TRANSMISSION (*U*) OF FRAME INTERIOR WALLS AND PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

WALL No.	TYPE OF WALL	SINGLE PARTITION (FINISH ON ONE SIDE OF STUDDING)	DOUBLE PARTITION (FINISHED ON BOTH SIDES OF STUDDING)			
			Air Space Between Studding	Flaked Gypsum Fill[b] Between Studding	Rock Wool Fill[b] Between Studding	½ In. Flexible Insulation Between Studding (One Air Space)
		A	B	C	D	E
53	**Wood Lath and Plaster** On Studding	0.62	0.34	0.11	0.071	0.21
54	**Metal Lath and Plaster**[c] On Studding	0.69	0.39	0.11	0.072	0.23
55	**Plaster Board (⅜ in.) and Plaster**[d] On Studding	0.61	0.34	0.10	0.071	0.21
56	**½ In. Rigid Insulation and Plaster**[d] On Studding	0.35	0.18	0.083	0.060	0.14
57	**1 In. Rigid Insulation and Plaster**[d] On Studding	0.23	0.12	0.066	0.051	0.097
58	**1½ In. Corkboard and Plaster**[d] On Studding	0.16	0.081	0.052	0.042	0.070
59	**2 In. Corkboard and Plaster**[d] On Studding	0.12	0.063	0.045	0.038	0.057

[a]Computed from factors given in Table 7.
[b]Thickness assumed 3⅝ in.
[c]Plaster on metal lath assumed ¾ in. thick.
[d]Plaster assumed ½ in. thick.

TABLE 12. COEFFICIENTS OF TRANSMISSION (*U*) OF MASONRY PARTITIONS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

No.	TYPE OF WALL	PLAIN WALLS (NO PLASTER)	WALLS PLASTERED ON ONE SIDE	WALLS PLASTERED ON BOTH SIDES
		A	B	C
60	**4-In. Hollow Clay Tile**	0.45	0.42	0.40
61	**4-In. Common Brick**	0.50	0.46	0.43
62	**4-In. Hollow Gypsum Tile**	0.30	0.28	0.27

[a]Computed from factors given in Table 7.

TABLE 13. COEFFICIENTS OF TRANSMISSION (U) OF FRAME CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

No.	TYPE OF CEILING	INSULATION BETWEEN JOISTS	TYPE OF FLOORING			
			COLUMN A. No flooring. COLUMN B. Yellow pine flooring[b] on joists. COLUMN C. Yellow pine flooring[b] on rigid insulation (½ in.) on joists. COLUMN D. Maple or oak flooring[c] on yellow pine sub-flooring[b] on joists.			
			A	B	C	D
1	No Ceiling	None	....	0.46	0.27	0.34
2	Metal Lath and Plaster (¾ in.)	None	0.69	0.30	0.21	0.25
3	Wood Lath and Plaster	None	0.62	0.28	0.20	0.24
4	Plaster Board (⅜ in.) and Plaster (½ in.)	None	0.61	0.28	0.20	0.24
5	Rigid Insulation (½ in.) and Plaster (½ in.)	None	0.35	0.21	0.16	0.18
6	Wood Lath and Plaster	Flexible[d] Insulation (½ in.)	0.23	0.16	0.13	0.14
7	Wood Lath and Plaster	Rigid Insulation[d] (½ in).	0.25	0.16	0.13	0.15
8	Wood Lath and Plaster	Flaked Gypsum Fill (2 in.)	0.17	0.13	0.11	0.12
9	Wood Lath and Plaster	Rock Wool Fill (2 in.)	0.12	0.098	0.086	0.092
10	Corkboard (1½ in.) and Plaster (½ in.)	None	0.16	0.12	0.10	0.11
11	Corkboard (2 in.) and Plaster (½ in.)	None	0.12	0.10	0.087	0.094

[a]Computed from factors given in Table 7.

[b]Thickness assumed to be 25/32 in.

[c]Thickness assumed to be 13/16 in.

[d]Based on one air space with no flooring, and two air spaces with flooring. The value of U will be the same if insulation is applied to under side of joists and separated from lath and plaster ceiling by 1-in. furring strips.

TABLE 14. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE CONSTRUCTION FLOORS AND CEILINGS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on still air (no wind) conditions on both sides.

TYPE OF FLOORING:
COLUMN A. No flooring (concrete bare)[b].
COLUMN B. Yellow pine flooring[c] on wood sleepers embedded in concrete[d].
COLUMN C. Maple or oak flooring[e] on yellow pine sub-flooring[c] on wood sleepers embedded in concrete.
COLUMN D. Tile or terrazzo[f] flooring on concrete.

No.	Thickness of Concrete (Inches)	Type of Ceiling	A	B	C	D
1	4	No Ceiling	0.65	0.40	0.31	0.61
2	6		0.59	0.37	0.30	0.56
3	8		0.53	0.35	0.28	0.51
4	10		0.49	0.33	0.27	0.47
5	4	½ in. Plaster Applied Directly to Under Side of Concrete	0.59	0.38	0.30	0.56
6	6		0.54	0.35	0.28	0.52
7	8		0.50	0.33	0.27	0.47
8	10		0.45	0.32	0.26	0.44
9	4	Suspended or Furred Metal Lath and Plaster (¾ in.) Ceiling	0.37	0.28	0.23	0.36
10	6		0.35	0.26	0.22	0.34
11	8		0.33	0.25	0.21	0.32
12	10		0.32	0.24	0.21	0.31
13	4	Suspended or Furred Ceiling of Plaster Board (⅜ in.) and Plaster (½ in.)	0.35	0.26	0.22	0.34
14	6		0.33	0.25	0.21	0.32
15	8		0.31	0.24	0.21	0.30
16	10		0.30	0.23	0.20	0.29
17	4	Suspended or Furred Ceiling of Rigid Insulation (½ in.) and Plaster (½ in.)	0.24	0.20	0.17	0.24
18	6		0.23	0.19	0.17	0.23
19	8		0.22	0.18	0.16	0.22
20	10		0.22	0.18	0.16	0.21
21	4	Plaster (½ in.) on Corkboard (1½ in.) Set in Cement Mortar (½ in.) on Concrete	0.15	0.13	0.12	0.14
22	6		0.14	0.13	0.12	0.14
23	8		0.14	0.12	0.11	0.14
24	10		0.14	0.12	0.11	0.14

[a] Computed from factors given in Table 7.
[b] The figures in COLUMN A may be used with sufficient accuracy for concrete floors covered with carpet or linoleum.
[c] Thickness of yellow pine flooring assumed to be 25/32 in.
[d] The figures in COLUMN B may be used with sufficient accuracy for maple or oak flooring[e] applied directly over the concrete on wood sleepers.
[e] Thickness of maple or oak flooring assumed to be 13/16 in.
[f] Thickness of tile or terrazzo assumed 1 in.

TABLE 15. COEFFICIENTS OF TRANSMISSION (U) OF CONCRETE FLOORS ON GROUND WITH VARIOUS TYPES OF FINISH FLOORING[a,e]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the ground and the air over the floor, and are based on still air (no wind) conditions.

TYPE OF FINISH FLOORING

COLUMN A. No flooring (concrete bare).
COLUMN B. Yellow pine flooring[b] on wood sleepers embedded in concrete.
COLUMN C. Maple or oak flooring[c] on yellow pine sub-flooring on wood sleepers embedded in concrete.
COLUMN D. Tile or terrazzo[d] on concrete.

No.	Thickness of Concrete (Inches)	Type and Thickness of Insulation	A	B	C	D
1	4	None	1.07	0.52	0.38	0.98
2	6		0.90	0.48	0.36	0.84
3	8		0.79	0.45	0.34	0.74
4	10		0.70	0.41	0.32	0.66
5	4	None[e]	0.66	0.40	0.31	0.63
6	8		0.54	0.35	0.28	0.52
7	4	1 in. Rigid Insulation[e]	0.22	0.18	0.16	0.22
8	8	1 in. Rigid Insulation[e]	0.21	0.17	0.15	0.20
9	4	2 in. Corkboard[e]	0.12	0.11	0.10	0.12
10	8	2 in. Corkboard[e]	0.12	0.11	0.10	0.12

[a]Computed from factors given in Table 7.

[b]Assumed $\frac{25}{32}$ in. thick.

[c]Assumed $\frac{13}{16}$ in. thick.

[d]Assumed 1 in. thick.

[e]The figures for Nos. 5 to 10, inclusive, include 3 in. cinder concrete placed directly on the ground. The insulation is applied between the cinder concrete and the stone concrete. Usually the insulation is protected on both sides by a waterproof membrane, but this is not considered in the calculations.

TABLE 16. COEFFICIENTS OF TRANSMISSION (*U*) OF VARIOUS TYPES OF FLAT ROOFS COVERED WITH BUILT-UP ROOFING[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between air at the two sides, and are based on an outside wind exposure of 15 mph.

WITHOUT CEILINGS—UNDERSIDE OF ROOF EXPOSED

COLUMN A. No insulation.
COLUMN B. Rigid insulation (½ in.).
COLUMN C. Rigid insulation (1 in.).
COLUMN D. Rigid insulation (1½ in.).
COLUMN E. Rigid insulation (2 in.).
COLUMN F. Corkboard (1 in.).
COLUMN G. Corkboard (1½ in.).
COLUMN H. Corkboard (2 in.).

WITH METAL LATH AND PLASTER CEILINGS[e]

COLUMN I. No insulation.
COLUMN J. Rigid insulation (½ in.).
COLUMN K. Rigid insulation (1 in.).
COLUMN L. Rigid insulation (1½ in.).
COLUMN M. Rigid insulation (2 in.).
COLUMN N. Corkboard (1 in.).
COLUMN O. Corkboard (1½ in.).
COLUMN P. Corkboard (2 in.).

No.	TYPE OF ROOF DECK	THICKNESS OF ROOF DECK (INCHES)	WITHOUT CEILINGS—UNDERSIDE OF ROOF EXPOSED								WITH METAL LATH AND PLASTER CEILINGS[e]							
			A	B	C	D	E	F	G	H	I	J	K	L	M	N	O	P
1	Precast Cement Tile	1⅝	0.85	0.37	0.24	0.18	0.14	0.22	0.16	0.13	0.43	0.26	0.19	0.15	0.12	0.18	0.14	0.11
2	Concrete	2	0.82	0.37	0.24	0.17	0.14	0.22	0.16	0.13	0.42	0.26	0.19	0.15	0.12	0.18	0.14	0.11
3	Concrete	4	0.72	0.34	0.23	0.17	0.13	0.21	0.16	0.12	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11
4	Concrete	6	0.64	0.33	0.22	0.16	0.13	0.21	0.15	0.12	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11
5	Wood	1[b]	0.49	0.28	0.20	0.15	0.12	0.19	0.14	0.12	0.32	0.21	0.16	0.13	0.11	0.15	0.12	0.10
6	Wood	1½[b]	0.37	0.24	0.18	0.14	0.11	0.17	0.13	0.11	0.26	0.19	0.15	0.12	0.10	0.14	0.11	0.095
7	Wood	2[b]	0.32	0.22	0.16	0.13	0.11	0.16	0.12	0.10	0.24	0.17	0.14	0.11	0.097	0.13	0.11	0.092
8	Wood	4[b]	0.23	0.17	0.14	0.11	0.096	0.13	0.11	0.091	0.18	0.14	0.12	0.10	0.087	0.11	0.096	0.082
9	Gypsum Fiber Concrete[c] (2 in.) on Plaster Board (⅜ in.)	2⅜	0.40	0.25	0.18	0.14	0.12	0.17	0.13	0.11	0.27	0.19	0.15	0.12	0.10	0.14	0.12	0.097
10	Gypsum Fiber Concrete[c] (3 in.) on Plaster Board (⅜ in.)	3⅜	0.32	0.22	0.16	0.13	0.11	0.15	0.12	0.10	0.23	0.17	0.14	0.11	0.097	0.13	0.11	0.091
11	Flat Metal Roofs[d]		0.95	0.39	0.25	0.18	0.14	0.23	0.17	0.13	0.46	0.27	0.19	0.15	0.12	0.18	0.14	0.11

[a]Computed from factors given in Table 7.

[b]Nominal thicknesses specified—actual thicknesses used in calculations.

[c]Gypsum fiber concrete—87½ per cent gypsum, 12½ per cent wood fiber.

[d]Coefficient of transmission of bare corrugated iron (no roofing) is 1.50 Btu per hour per square foot of projected area per degree Fahrenheit difference in temperature, based on an outside wind velocity of 15 mph.

[e]These coefficients may be used with sufficient accuracy for wood lath and plaster, or plaster board and plaster ceilings. It is assumed that there is an air space between the under side of the roof deck and the upper side of the ceiling.

TABLE 17. COEFFICIENTS OF TRANSMISSION (U) OF PITCHED ROOFS[a]

Coefficients are expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air on the two sides, and are based on an outside wind velocity of 15 mph.

TYPE OF CEILING (APPLIED DIRECTLY TO ROOF RAFTERS)

COLUMN A. No ceiling (rafters exposed).
COLUMN B. Metal lath and plaster (3/4 in.).
COLUMN C. Plaster board (3/8 in.) and plaster (1/2 in.).
COLUMN D. Wood lath and plaster.
COLUMN E. Rigid insulation (1/2 in.).
COLUMN F. Rigid insulation (1/2 in.) and plaster (1/2 in.).
COLUMN G. Rigid insulation (1 in.) and plaster (1/2 in.).
COLUMN H. Corkboard (1 1/2 in.) and plaster (1/2 in.).
COLUMN I. Corkboard (2 in.) and plaster (1/2 in.).

No.	TYPE OF ROOFING AND ROOF SHEATHING	INSULATION BETWEEN ROOF RAFTERS	A	B	C	D	E	F	G	H	I
1	Wood Shingles on Wood Strips[b]	None	0.48	0.30	0.29	0.29	0.22	0.21	0.16	0.12	0.10
2		1/2 in. Flexible[c]	....	0.17	0.16	0.16	0.14	0.13	0.11	0.091	0.079
3		1 in. Flexible[c]	....	0.13	0.12	0.12	0.11	0.11	0.092	0.078	0.069
4		3 5/8 in. Flaked Gypsum[e]	....	0.097	0.096	0.096	0.086	0.085	0.076	0.066	0.059
5		3 5/8 in. Rock Wool[e]	....	0.065	0.065	0.065	0.062	0.060	0.055	0.050	0.046
6	Asphalt Shingles, Rigid Asbestos Shingles, Composition Roofing, or Slate or Tile Roofing[d] on Wood Sheathing[f]	None	0.56	0.34	0.32	0.32	0.24	0.23	0.17	0.13	0.11
7		1/2 in. Flexible[c]	....	0.18	0.17	0.17	0.14	0.14	0.12	0.094	0.089
8		1 in. Flexible[c]	....	0.13	0.13	0.13	0.11	0.11	0.095	0.080	0.071
9		3 5/8 in. Flaked Gypsum	....	0.10	0.10	0.10	0.092	0.091	0.080	0.069	0.062
10		3 5/8 in. Rock Wool[e]	....	0.071	0.070	0.070	0.065	0.064	0.059	0.053	0.048

[a]Computed from factors given in Table 7. Nos. 6 to 10, inclusive, based on 1/2 in. thick slate.
[b]Based on 1 in. by 4 in. strips spaced 2 in.
[c]Figures based on two air spaces. Insulation may also be applied to under side of roof rafters with furring strips between.
[d]Roofing felt between roof sheathing and slate or tile neglected in calculations.
[e]Assumed 3 5/8 in. thick based on the actual width of 2 in. by 4 in. rafters.
[f]Sheathing assumed 25/32 in. thick.

Basements and Unheated Rooms

The heat loss through floors into basements and into unheated rooms kept closed may be computed by assuming a temperature for these rooms of 32 F.

Additional information on the inside and outside temperatures to be used in heat loss calculations is given in Chapter 7.

TABLE 18. COEFFICIENTS OF TRANSMISSION (U) OF DOORS, WINDOWS AND SKYLIGHTS

Coefficients are based on a wind exposure of 15 mph, and are expressed in Btu per hour, per square foot, per degree Fahrenheit difference in temperature between the air inside and outside of the door, window or skylight.

A. Windows and Skylights

	U
Single	1.13[a-c]
Double	0.45[a]
Triple	0.281[a]

B. Solid Wood Doors[b-c]

Nominal Thickness Inches	Actual Thickness Inches	U
1	25/32	0.69
1¼	1 1/16	0.59
1½	1 5/16	0.52
1¾	1⅜	0.51
2	1⅝	0.46
2½	2⅛	0.38
3	2⅝	0.33

[a]See *Heating, Ventilating and Air Conditioning*, by Harding and Willard, revised edition, 1932.
[b]Computed using C = 1.15 for wood; f_i = 1.65 and f_o = 6.0.
[c]It is sufficiently accurate to use the same coefficient of transmission for doors containing thin wood panels, as that of single panes of glass, namely, 1.13 Btu per hour per square foot per degree difference between inside and outside air temperature.

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Chapter 6

AIR FILTRATION

Causes of Air Leakage, Air Leakage Through Walls, Window Leakage, Wind Velocity to be Selected, Crack used for Computations, Multi-Story Buildings, Heat Equivalent of Air Entering by Infiltration

INFILTRATION (or exfiltration) losses are those resulting from the displacement of heated air in a building by unheated outside air, the interchange taking place through various apertures in the building, such as cracks around doors and windows, fireplaces and chimneys. This leakage of air must be considered in heating and cooling calculations. (See Chapters 7 and 8).

CAUSES OF AIR LEAKAGE

A building is a shell in which the internal pressure in general is not in equilibrium with the external pressure. At some places, the latter is greater than the former, and inflow will occur in such regions through any openings in the wall whether large or small. At other places the reverse is true. The internal pressure automatically assumes a value to correspond to the requirement that outflow must equal inflow. The agencies that cause leakage are the natural forces of wind and temperature difference, and those produced by fans if any are used. A consideration of all factors involved in causing pressure difference at various places about a building would present an exceedingly complex problem.

In tall single story buildings, the *chimney effect* caused by the inside-outside temperature difference becomes a factor of importance. Even in multi-story buildings it usually is not possible to isolate the several floors completely, and chimney effect is operative to a considerable degree, tending to force air in at the lower levels and out at the upper.

Since the full force of the wind usually is not the effective pressure differential, owing to back pressure built up within the building, the actual amount of infiltration (cubic feet) is assumed to be 80 per cent of that determined in laboratory experiments.

AIR LEAKAGE THROUGH WALLS

Table 1[1] gives data on infiltration through brick and frame walls. The brick walls listed in this table are walls which show poor workmanship

[1]Air Infiltration through Various Types of Brick Wall Construction, by Larson, Nelson and Braatz (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

and which are constructed of porous brick and lime mortar. For good workmanship, the leakage through hard brick walls with cement-lime mortar does not exceed one-third the values given. These tests indicate that plastering reduces the leakage by about 96 per cent; a heavy coat of cold water paint, 50 per cent; and 3 coats of oil paint carefully applied, 28 per cent. The infiltration through walls ranges from 6 to 25 per cent of that through windows and doors in a 10-story office building, with imperfect sealing of plaster at the baseboards of the rooms. With perfect sealing the range is from 0.5 to 2.7 per cent or a practically negligible quantity, which indicates the importance of good workmanship in proper

TABLE 1. INFILTRATION THROUGH WALLS

Expressed in cubic feet per square foot per hour[a]

Type of Wall		Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
8½ in. Brick Wall	Plain	1.75	4.20	7.85	12.2	18.6	22.9
	Plastered	0.017	0.037	0.066	0.107	0.161	0.236
13 in. Brick Wall	Plain	1.44	3.92	7.48	11.6	16.3	21.2
	Plastered	0.005	0.013	0.025	0.043	0.067	0.097
Frame Wall, with lath and plaster[b]		0.03	0.07	0.13	0.18	0.23	0.26
Frame Wall, with lath and plaster[c]		0.02	0.05	0.10	0.15	0.20	0.25

[a]The values in this table are 20 per cent less than test values to allow for building up of pressure in rooms and are based on test data reported in A.S.H.V.E. research papers entitled, Air Infiltration Through Various Types of Brick Wall Construction, and Air Infiltration Through Various Types of Wood Frame Construction. (See References on p. 102).

[b]Wall construction: Bevel siding painted, No. 1 common butt-edged sheathing, building paper, wood lath and 3 coats gypsum plaster.

[c]Wall construction: Cedar shingle, shiplap sheathing, building paper, wood lath and 3 coats gypsum plaster.

sealing at the baseboard. It will be noted from Table 1, that the infiltration through properly plastered walls can be neglected.

The value of building paper when applied between sheathing and shingles is indicated by Fig. 1, which represents the effect on outside construction only without lath and plaster. The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them.

The amount of infiltration that may be expected through simple walls used in farm and other shelter buildings, is shown in Fig. 2. The infil-

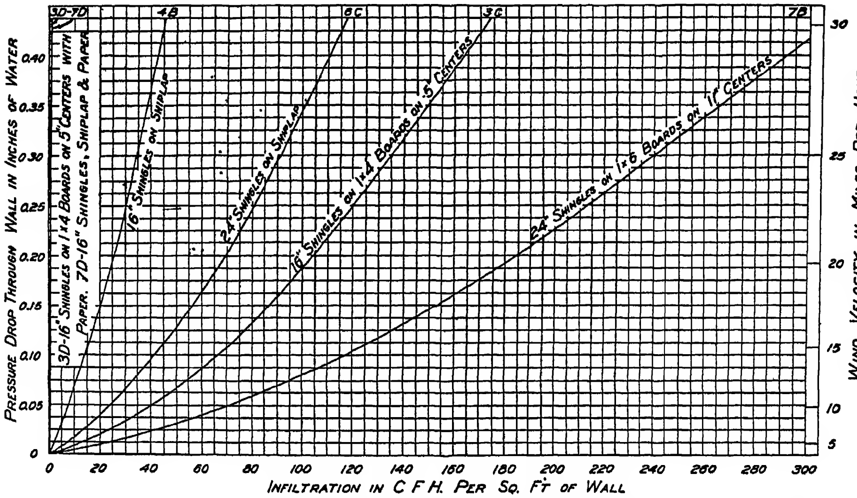

FIG. 1. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

tration there indicated is that determined in the laboratory and should be multiplied by the factor 0.80 to give proper working values.

WINDOW LEAKAGE

The amount of infiltration in cubic feet per hour per foot of crack for various types of windows is given in Table 2. The distinction between *crack* and *clearance* of a wood sash is shown by Fig. 3. *For window and*

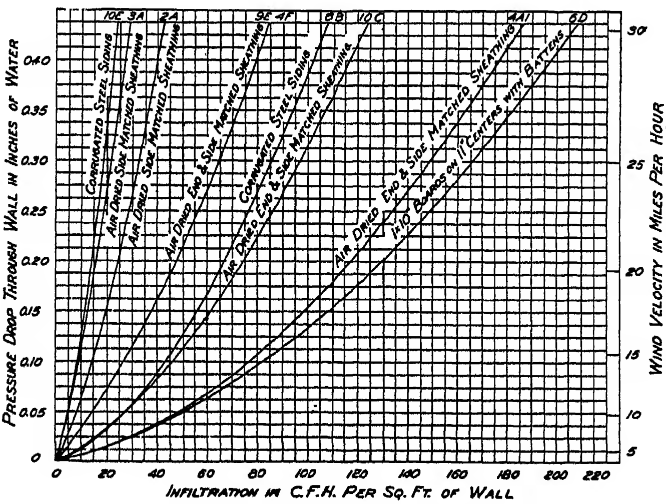

FIG. 2. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

Table 2. Infiltration Through Windows

Expressed in Cubic Feet per Foot of Crack per Hour[a]

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked[b]	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked[b]	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction[b]	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, 1/16-in. crack and 3/64-in. clearance[c]. Includes wood frame leakage[d]	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped[d]	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, 3/32-in. crack and 3/32-in. clearance[e]. Includes wood frame leakage[d].	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped[d]	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows[f]	Non-weatherstripped, locked	20	45	70	96	125	154
	Non-weatherstripped, unlocked	20	47	74	104	137	170
	Weatherstripped, unlocked	6	19	32	46	60	76
Rolled Section Steel Sash Windows[k]	Industrial pivoted,[g] 1/16-in. crack	52	108	176	244	304	372
	Architectural projected,[h] 3/64-in. crack	20	52	88	116	152	208
	Residential casement,[i] 1/32-in. crack	14	32	52	76	100	128
	Heavy casement section, projected,[j] 1/32-in. crack	8	24	38	54	72	96
Hollow Metal, vertically pivoted window[f]		30	88	145	186	221	242

[a]The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms, and are based on test data reported in the papers listed at the end of this chapter.

[b]The values given for frame leakage are per foot of sash perimeter as determined for double-hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

[c]The fit of the average double-hung wood window was determined as 1/16-in. crack and 3/64-in. clearance by measurements on approximately 600 windows under heating season conditions.

[d]The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction assuming a 50 per cent efficiency of frame calking.

[e]A 3/32-in. crack and clearance represents a poorly fitted window, much poorer than average.

[f]Windows tested in place in building.

[g]Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

[h]Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms.

[i]Of same design and section shapes as so-called *heavy section casement* but of lighter weight.

[j]Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening.

[k]With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With 3/64-in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

sash leakage, the length of crack in double-hung windows is equal to the perimeter of sash plus length of meeting rail. For steel sash the length of crack is the aggregate perimeter of the movable or ventilating sections plus the linear feet of sash section in contact with steel work (at a different leakage rate) at mullions. The crack length for *frame windows* (when frame is not calked) is the perimeter of the frame. Steel sash frame properly grouted with cement mortar into brickwork or concrete is not to be counted as crack. The value of storm sash is shown by the curves of Figs. 4 and 5. A study of the curves leads to the conclusion that a storm sash is of little value in reducing infiltration when applied to a well fitted window, but that a reduction of 50 per cent might be expected when storm sash is applied to a poorly fitted or loose window. Infiltration through door cracks may be assumed to be twice that of window cracks.

WIND VELOCITY TO BE CHOSEN

Although all authorities do not agree upon the value of the wind velocity that should be chosen for any given locality, it is common engineering

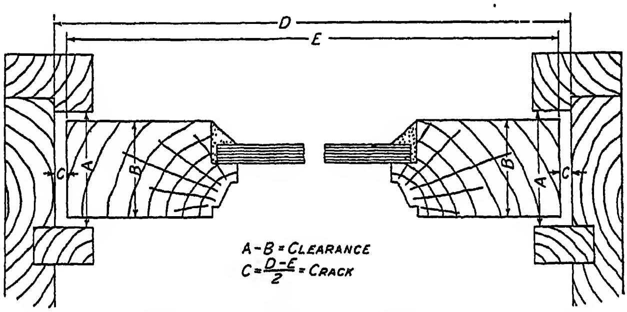

FIG. 3. DIAGRAM ILLUSTRATING CRACK AND CLEARANCE

practice to use the average wind velocity during the three coldest months of the year. Until this point is definitely established the practice of using average values will be followed. Average wind velocities for the months of December, January and February for various cities in the United States and Canada are given in Table 2, Chapter 7.

In considering both the transmission and infiltration losses, the more exact procedure would be to select the outside temperature and the wind velocity corresponding thereto, based on Weather Bureau records, which would result in the maximum heat demand. Since the proportion of transmission and infiltration losses varies with the construction and is different for every building, the proper combination of temperature and wind velocity to be selected would be different for every type of building, even in the same locality. Furthermore, such a procedure would necessitate a laborious cut-and-try process in every case in order to determine the worst combination of conditions for the building under consideration. It would also be necessary to consider heat lag due to heat capacity in the case of heavy masonry walls, and other factors, to arrive at the most accurate solution of the problem. Although heat capacity should be considered wherever possible, it is seldom possible to accurately determine the

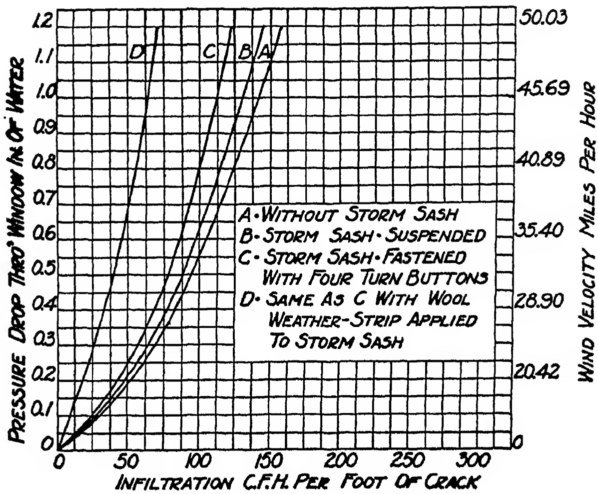

Fig. 4. Infiltration through Sash Perimeter of Window with and without Storm Sash—5/64-in. Crack and 1/32-in. Clearance

worst combination of outside temperature and wind velocity for a given building and locality. The usual procedure, as already explained, is to select an outside temperature based on the lowest on record and the average wind velocity during the months of December, January and February.

The direction of prevailing winds may usually be included within an angle of about 90 deg. The windows that are to be figured for prevailing

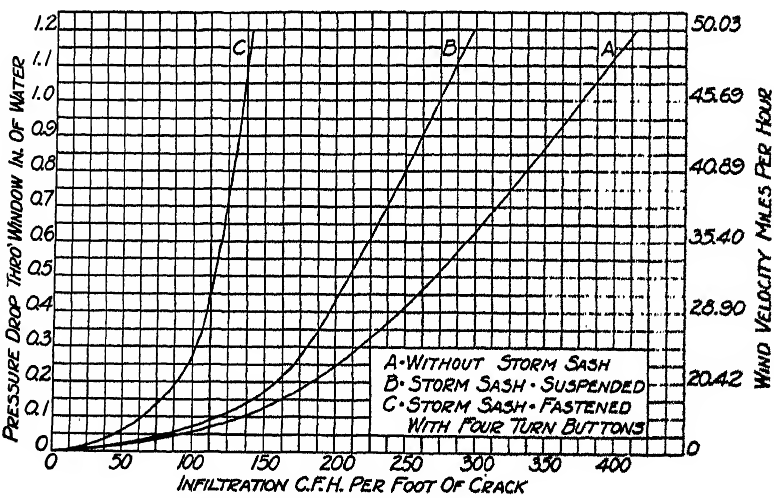

Fig. 5. Infiltration through Sash Perimeter of Window with and without Storm Sash—1/8-in. Crack and 1/8-in. Clearance

and non-prevailing winds will ordinarily each occupy about one-half the perimeter of the structure, the proportion varying to a considerable extent with the plan of the structure. (See discussion of wind movement in Chapter 4).

CRACK USED FOR COMPUTATIONS

In no case should the amount of crack used for computation be less than half of the total crack in the outside walls of the room. Thus, in a room with one exposed wall, take all the crack; with two exposed walls, take the wall having the most crack; and with three or four exposed walls, take the wall having the most crack; but in no case take less than half the total crack. For a building having no partitions, whatever wind enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, take one-half the total crack for computing each side and end of the building.

The amount of air leakage is sometimes roughly estimated by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room, as indicated in Table 3.

TABLE 3. AIR CHANGES TAKING PLACE UNDER AVERAGE CONDITIONS EXCLUSIVE OF AIR PROVIDED FOR VENTILATION

KIND OF ROOM OR BUILDING	NUMBER OF AIR CHANGES TAKING PLACE PER HOUR
Rooms, 1 side exposed	1
Rooms, 2 sides exposed	1½
Rooms, 3 sides exposed	2
Rooms, 4 sides exposed	2
Rooms with no windows or outside doors	½ to ¾
Entrance Halls	2 to 3
Reception Halls	2
Living Rooms	1 to 2
Dining Rooms	1 to 2
Bath Rooms	2
Drug Stores	2 to 3
Clothing Stores	1
Churches, Factories, Lofts, etc.	½ to 3

MULTI-STORY BUILDINGS

In tall buildings, infiltration may be considerably influenced by temperature difference or chimney effect which will operate to produce a head that will add to the effect of the wind at lower levels and subtract from it at higher levels. On the other hand, the wind velocity at lower levels may be somewhat abated by surrounding obstructions. Furthermore, the chimney effect is reduced in multi-story buildings by the partial isolation of floors preventing free upward movement, so that wind and temperature difference may seldom cooperate to the fullest extent. Making the rough assumption that the *neutral zone* is located at mid-height of a building, and that the temperature difference is 70 F, the

following formulae may be used to determine an equivalent wind velocity to be used in connection with Tables 1 and 2 that will allow for both wind velocity and temperature difference:

$$M_e = \sqrt{M^2 - 1.75\,a} \qquad (1)$$

$$M_e = \sqrt{M^2 + 1.75\,b} \qquad (2)$$

where

M_e = equivalent wind velocity to be used in conjunction with Tables 1 and 2.
M = wind velocity upon which infiltration would be determined if temperature difference were disregarded.
a = distance of windows under consideration from mid-height of building if *above* mid-height.
b = distance if *below* mid-height.

The coefficient 1.75 allows for about one-half the temperature difference head.

Example 1. If M = 15. the equivalent wind velocity at a height of 150 ft from the ground for a building 180 ft high would be

$$M_e = \sqrt{15^2 - 1.75 \times 60} = 11 \text{ mph}$$

For a window on the ground floor:

$$M_e = \sqrt{15^2 + 1.75 \times 90} = 19.6 \text{ mph}$$

For buildings of unusual height, Equation 1 would indicate negative infiltration at the highest stories, which condition may, at times, actually exist, although probably greater wind velocities should be figured at such extremely high levels.

Sealing of Vertical Openings[2]

In tall, multi-story buildings, every effort should be made to seal off vertical openings such as stair-wells and elevator shafts from the remainder of the building. Stair-wells should be equipped with self-closing doors, and in exceptionally high buildings, should be closed off into sections of not over 10 floors each. Plaster cracks should be filled. Elevator enclosures should be tight and solid doors should be used.

If the sealing of the vertical openings is made effective, no allowance need be made for the chimney effect. Instead the greater wind movement at the high altitudes makes it advisable to install additional heating surface on the upper floors above the level of neighboring buildings, this additional surface being increased as the height is increased. One arbitrary rule is to increase the heating surface on floors above neighboring buildings by an amount ranging from 5 per cent to 20 per cent. This extra heating surface is required only on the windward side and on windy days, and hence automatic temperature control is especially desirable with such installations.

Heating Surface for Stair-Wells[2]

In stair-wells that are open through many floor levels although closed off from the remainder of each floor by doors and partitions, the strati-

[2]See Flue Action in Tall Buildings, by H. L. Alt (*Heating, Piping and Air Conditioning*, May, 1932).

fication of air makes it advisable to increase the amount of heating surface at the lower levels and to decrease the amount at higher levels even to the point of omitting all heating surface on the top several floor levels. One rule is to calculate the heating surface of the entire stair-well in the usual way and to place 50 per cent of this in the bottom third, the normal amount in the middle third and the balance in the top third.

HEAT EQUIVALENT OF AIR ENTERING BY INFILTRATION

The heat required to warm cold, outside air, which enters a room by infiltration, to the temperature of the room is given by the following equation:

$$H_i = 0.24\, Q\, d\, (t - t_o) \tag{3}$$

where

H_i = Btu per hour required for heating air leaking into building from outside temperature t_o to inside temperature t.

Q = cubic feet of air entering per hour at inside temperature t.

d = density (pounds per cubic foot) of air at inside temperature t.

t = inside temperature at the proper level.

t_o = outside air temperature for which heating system is designed.

0.24 = specific heat of air.

It is sufficiently accurate to take d = 0.075 lb, in which case the equation reduces to

$$H_i = 0.018\, Q\, (t - t_o) \tag{4}$$

While a heating reserve must be provided to warm inleaking air on the windward side of a building, this does not necessarily mean that the heating plant must be provided with a reserve capacity, since the inleaking air, warmed at once by adequate heating surface in exposed rooms, will move transversely and upwardly through the building, thus relieving other radiators of a part of their load. The actual loss of heat of a building caused by infiltration is not to be confused with the necessity for providing additional heating capacity for a given space. Infiltration is a disturbing factor in the heating of a building, and its maximum effect (maximum in the sense of an average of wind velocity peaks during the heating season above some reasonably chosen minimum) must be met by a properly distributed reserve of heating capacity, which reserve, however, is not in use at all places at the same time, nor in any one place at all times.

Example 2. A 12 ft by 18 ft room with a ceiling height of 10 ft contains three 2 ft-8 in. by 5 ft-6 in. plain double-hung wood windows with $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance. Assume a wind velocity of 20 mph and a temperature difference of 75 F. Neglecting chimney effect, what is the maximum heat loss due to infiltration?

Solution. From Table 2, the leakage per foot of crack is 59.3 cfh. Length of crack for the three windows is 57 ft. The infiltration (Q) is equal to 59.3 × 57 or 3380 cfh, and the additional heat loss (maximum) due to infiltration is equal to 0.018 × 3380 × 75 or 4560 Btu per hour. (Since the room has a volume of 2160 cu ft, the air changes would be $\frac{3380}{2160}$ or 1.57 per hour).

Example 3. What is the probable inleakage of air for a room with three windows on the first floor, if the wind velocity is 15 mph? The building is 90 ft high and is equipped with architectural projected steel windows with two ventilators each with a total perimeter of 24 ft of 3/64-in. crack. Further, find the probable inleakage in similar rooms on the top floor and on the third floor 30 ft from the ground.

Solution. On the first floor the equivalent wind velocity is $M_e = \sqrt{15^2 + 1.75 \times 45}$ or 17.5 mph. Leakage for this type of window is given in Table 2 (interpolating) as 102 cu ft per foot of crack per hour. The total leakage rate (Q) is equal to $3 \times 24 \times 102$ or 7344 cu ft per hour.

For a similar room on the top floor, with the same exposure, the equivalent wind velocity is $M_e = \sqrt{15^2 - 1.75 \times 45}$ or 12 mph for which the leakage is 66.4 cu ft per ft of crack per hour. The total inleakage of air into the room (Q) is equal to $3 \times 24 \times 66.4$ or 4780 cu ft per hour.

For a similar room 30 ft above the ground, $M_e = \sqrt{15^2 + 1.75 \times 15}$ or 16 mph for which the unit crack leakage will be 93.6 cu ft per hour, and the total leakage for the room will be 6740 cu ft per hour.

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Chapter 7

THE HEATING LOAD

Factors Governing Heat Demand, Procedure, Temperatures, Wind Movement, Heat Sources Other Than Heating Plant, Example, Condensation

IN the design of any type of heating system, the maximum probable heat demand must be accurately estimated in order that the apparatus installed shall be of sufficient capacity to maintain the desired temperature at all times. The factors which govern this maximum heat demand—most of which are seldom, if ever, in equilibrium—include the following:

1. Outside temperature. 2. Rain or snow. 3. Sunshine or cloudiness. 4. Wind velocity.	*Outside Conditions (The Weather)*
5. Heat transmission of exposed parts of building. 6. Infiltration of air through cracks, crevices and open doors and windows. 7. Heat capacity of materials. 8. Rate of absorption of solar radiation by exposed materials.	*Building Construction*
9. Inside temperatures. 10. Stratification of air. 11. Type of heating system. 12. Ventilation requirements. 13. Period and nature of occupancy. 14. Temperature regulation.	*Inside Conditions*

The *inside conditions* vary from time to time, the physical properties of the *building construction* may change with age, and the *outside conditions* are changing constantly. Just what the worst combination of all of these variable factors is likely to be in any particular case is therefore conjectural. Because of the nature of the problem, extreme precision in estimating heat losses at any time is very unlikely.

The procedure to be followed in determining the heat loss from any building can be divided into seven consecutive steps, as follows:

1. Determine on the inside air temperature, at the breathing line, or the 30-in. line, which is to be maintained in the building during the coldest weather. (See Table 1).

2. Determine on an outside air temperature for design purposes, based on the minimum temperatures recorded in the locality in question, which will provide for all but the most severe weather conditions. Such conditions as may exist for only a few consecutive hours are readily taken care of by the heat capacity of the building itself. (See Table 2).

3. Select or compute the heat transmission coefficients for outside walls and glass;

also for inside walls, floors, or top-floor ceilings, if these are next to unheated space; include roof if next to heated space. (See Chapter 5).

4. Measure up net outside wall, glass and roof next to heated spaces, as well as any cold walls, floors or ceilings next to unheated space. Such measurements are made from building plans, or from the actual building.

5. Compute the heat transmission losses for each kind of wall, glass, floor, ceiling and roof in the building by multiplying the heat transmission coefficient in each case by the area of the surface in square feet and the temperature difference between the inside and outside air. (See items 1 and 2).

6. Select unit values and compute the heat equivalent of the infiltration of cold air taking place around outside doors and windows. These unit values depend on the kind or width of crack and wind velocity, and when multiplied by the length of crack and the temperature difference between the inside and outside air, the result expresses the heat required to warm up the cold air leaking into the building per hour. (See Chapter 6).

7. The sum of the heat losses by transmission (item 5) through the outside wall and glass, as well as through any cold floors, ceilings or roof, plus the heat equivalent (item 6) of the cold air entering by infiltration represents the total heat loss equivalent for any building.

TABLE 1. WINTER INSIDE DRY-BULB TEMPERATURES USUALLY SPECIFIED*

TYPE OF BUILDING	DEG FAHR	TYPE OF BUILDING	DEG FAHR
SCHOOLS		THEATERS—	
Class Rooms	70–72	Seating Space	68–72
Assembly Rooms	68–72	Lounge Rooms	68–72
Gymnasiums	55–65	Toilets	68
Toilets and Baths	70		
Wardrobe and Locker Rooms	65–68	HOTELS—	
Kitchens	66	Bedrooms and Baths	70
Dining and Lunch Rooms	65–70	Dining Rooms	70
Playrooms	60–65	Kitchens and Laundries	66
Natatoriums	75	Ball Rooms	65–68
		Toilets and Service Rooms	68
HOSPITALS—			
Private Rooms	70–72	HOMES	70–72
Private Rooms (surgical)	70–80	STORES	65–68
Operating Rooms	70–95	PUBLIC BUILDINGS	68–72
Wards	68	WARM AIR BATHS	120
Kitchens and Laundries	66	STEAM BATHS	110
Toilets	68	FACTORIES AND MACHINE SHOPS	60–65
Bathrooms	70–80	FOUNDRIES AND BOILER SHOPS	50–60
		PAINT SHOPS	80

*The most comfortable dry-bulb temperature to be maintained depends on the relative humidity and air motion. These three factors considered together constitute what is termed the *effective temperature*. See Chapter 2.

Item 7 represents the heat losses after the building is heated and under stable operating conditions in coldest weather. Additional heat is required for raising the temperature of the air, the building materials and the material contents of the building to specified standard inside temperature.

The rate at which this additional heat is required depends upon the heat capacity of the structure and its material contents and upon the time in which these are to be heated.

This additional heat may be figured and allowed for as conditions require, but inasmuch as the heating system proportioned for taking care

of the heat losses will usually have a capacity about 100 per cent greater than that required for average winter weather, and inasmuch as most buildings may either be continuously heated or more time be allowed for heating-up during the few minimum temperature days, no allowance is made except in the size of boilers or furnaces.

INSIDE TEMPERATURES

The inside air temperature which must be maintained within a building and which should always be stated in the heating specifications is understood to be the dry-bulb temperature at the breathing line, 5 ft above the floor, or the 30-in. line, and not less than 3 ft from the outside walls. Inside air temperatures, usually specified, vary in accordance with the use to which the building is to be put and Table 1 presents values which conform with good practice.

The proper dry-bulb temperature to be maintained depends upon the relative humidity and air motion, as explained in Chapter 2. In other words, a person may feel warm or cool at the same dry-bulb temperature, depending on the relative humidity and air motion. The optimum winter *effective temperature* for sedentary persons, as determined at the A.S.H. V.E. Research Laboratory, is 66 deg.[1]

According to Fig. 2, Chapter 2, for so-called still air conditions, a relative humidity of approximately 50 per cent is required to produce an effective temperature of 66 deg when the dry-bulb temperature is 70 F. However, even where provision is made for artificial humidification, the relative humidity is seldom maintained higher than 40 per cent during the extremely cold weather, and where no provision is made for humidification, the relative humidity may be 20 per cent or less. Consequently, in using the figures given in Table 1, consideration should be given to whether provision is to be made for humidification, and if so, the actual relative humidity to be maintained.

Temperature at Proper Level: In making the actual heat-loss computations, however, for the various rooms in a building it is often necessary to modify the temperatures given in Table 1 so that the air temperature at the proper level will be used. By *air temperature at the proper level* is meant, in the case of walls, the air temperature at the mean height between floor and ceiling; in the case of glass, the air temperature at the mean height of the glass; in the case of roof or ceiling, the air temperature at the mean height of the roof or ceiling above the floor of the heated room; and in the case of floors, the air temperature at the floor level. In the case of heated spaces adjacent to unheated spaces, it will usually be sufficient to assume the temperature in such spaces as the mean between the temperature of the inside heated spaces and the outside air temperature, excepting where the combined heat transmission coefficient of the roof and ceiling can be used, in which case the usual inside and outside temperatures should be applied. (See discussion regarding the use of combined coefficients of pitched roofs, unheated attics and top-floor ceilings on p. 91).

[1]See Chapter 2, p. 27.

High Ceilings: Research data concerning stratification of air in buildings are lacking, but in general it may be said that where the increase in temperature is due to the natural tendency of the warmer or less dense air to rise, as where a direct radiation system is installed, the temperature of the air at the ceiling increases with the ceiling height. The relation, however, is not a straight-line function, as the amount of increase per foot of height apparently decreases as the height of the ceiling increases, according to present available information.

It is the common practice of engineers to allow a change in temperature of 2 per cent per foot of height above the breathing line in determining the probable air temperature at any given level for a direct radiation system, and this value is, no doubt, sufficiently accurate in most cases, although it is not probable that this rule applies to heights above 20 ft.

With certain types of heating and ventilating systems, which tend to oppose the natural tendency of warm air to rise, the temperature differential between floor and ceiling can be greatly reduced. These include unit heaters, fan-furnace heaters, and the various types of mechanical ventilating systems. The amount of reduction is problematical in certain instances, as it depends upon many factors such as location of heaters, air temperature, and direction and velocity of air discharge. In some cases it has been possible to reduce the temperature between the floor and ceiling to a few degrees, whereas, in other cases, the temperature at the ceiling has actually been increased because of improper design, installation or operation of equipment. So much depends upon the factors enumerated, that it is not advisable to allow less than 1 per cent per foot (and usually more) above the breathing line in arriving at the air temperature at any given level for any of these types of heating and ventilating systems, unless the manufacturers are willing to guarantee that the particular type of equipment under consideration will maintain a smaller temperature differential for the specific conditions involved.

Temperature at Floor Level: In determining mean air temperatures just above floors which are next to ground or unheated spaces, a temperature 5 deg lower than the breathing-line temperature may be used, provided the breathing-line temperature is not less than 55 F.

OUTSIDE TEMPERATURES

The outside temperature used in computing the heat loss from a building is seldom taken as the lowest temperature ever recorded in a given locality. Such temperatures are usually of short duration and are rarely repeated in successive years. It is therefore evident that a temperature somewhat higher than the lowest on record may be properly assumed in making the heat-loss computations.

The outside temperature to be assumed in the design of any heating system must not be more than 15 deg above the lowest recorded temperature as reported by the Weather Bureau during the preceding 10 years for the locality in which the heating system is to be installed. In the case of massive and well-insulated buildings in localities where the minimum does not prevail for more than a few hours, it is possible that more than 15 deg above the minimum may be allowed, due primarily to the *fly-wheel* effect of the heat capacity of the structure. The outside

temperature assumed and used in the design should always be stated in the heating specifications. Table 2 lists the coldest dry-bulb temperatures ever recorded by the Weather Bureau.

If Weather Bureau reports are not available for the locality in question, then the reports for the station nearest to this locality are to be used, unless some other temperature is specifically stated in the specifications. In computing the average heat transmission losses for the heating season in the United States the average outside temperature from October 1 to May 1 should be used.

WIND MOVEMENT

The effect of wind on the heating requirements of any building should be given consideration under two heads:

1. Wind movement increases the heat transmission of walls, glass, and roof, affecting poor walls to a much greater extent than good walls.
2. Wind movement materially increases the infiltration (inleakage) of cold air through the cracks around doors and windows, and even through the building materials themselves, if such materials are at all porous.

It is entirely possible that a building may require more heat on a windy day with a moderately low outside temperature than on a quiet day with a much lower outside temperature. It will, therefore, be evident that the wind movement in any locality must be given careful consideration in computing the probable heating requirements of a building, and for the purposes of calculation, not less than the average wind movement in any locality during December, January and February should always be provided for in computing (1) the heat transmission of a building, and (2) the heat required to take care of the infiltration of outside air.

The first condition is readily taken care of, as explained in Chapter 5, by using a surface coefficient f_o for the outside wall surface, which is based on the proper wind velocity. In case specific data are lacking for any given locality, it is sufficiently accurate to use an average wind velocity of approximately 15 mph which is the velocity upon which the heat transmission coefficient tables in Chapter 5 are based.

In a similar manner, the heat allowance for infiltration through cracks and walls (Tables 1 and 2, Chapter 6) must be based on the proper wind velocity for a given locality, as explained in Chapter 6. In the case of *tall buildings*, special attention must be given to infiltration factors, as also explained in Chapter 6.

HEAT FROM SOURCES OTHER THAN HEATING PLANT

The heat supplied by persons, lights, motors and machinery should always be ascertained in the case of theaters, assembly halls, and industrial plants, but allowances for such heat sources must be made only after careful consideration of all local conditions. In many cases, these heat sources should not be allowed to affect the size of the installation at all, although they may have a marked effect on the operation and control of the system. In general, it is safe to say that where audiences are involved, the heating installation must have sufficient capacity to bring the building up to the stipulated inside temperature before the audience

Table 2. Climatic Conditions Compiled from Weather Bureau Records

Col. A	Col. B	Col. C	Col. D	Col. E	Col. F
State	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hr	Direction of Prevailing Wind, Dec., Jan., Feb.
Ala.	Mobile	57.7	−1	8.3	N
	Birmingham	53.9	−10	8.6	N
Ariz.	Phoenix	59.5	16	3.9	E
	Flagstaff	34.9	−25	6.7	SW
Ark.	Fort Smith	49.5	−15	8.0	E
	Little Rock	51.6	−12	9.9	NW
Cal.	San Francisco	54.3	29	....	N
	Los Angeles	58.6	28	....	NE
Colo.	Denver	39.3	−29	7.4	S
	Grand Junction	39.2	−16	5.6	SE
Conn.	New Haven	38.0	−14	9.3	N
D. C.	Washington	43.2	−15	7.3	NW
Fla.	Jacksonville	61.9	10	8.2	NE
Ga.	Atlanta	51.4	−8	11.8	NW
	Savannah	58.4	8	8.3	NW
Idaho	Lewiston	42.5	−13	4.7	E
	Pocatello	36.4	−20	9.3	SE
Ill.	Chicago	36.4	−23	17.0	SW
	Springfield	39.9	−24	10.2	NW
Ind.	Indianapolis	40.2	−25	11.8	S
	Evansville	44.1	−15	8.4	S
Iowa	Dubuque	33.9	−32	6.1	NW
	Sioux City	32.1	−35	12.2	NW
Kan.	Concordia	38.9	−25	7.3	N
	Dodge City	40.2	−26	10.4	NW
Ky.	Louisville	45.2	−20	9.3	SW
La.	New Orleans	61.5	7	9.6	N
	Shreveport	56.2	−5	7.7	SE
Me.	Eastport	31.1	−23	13.8	W
	Portland	33.6	−17	10.1	NW
Md.	Baltimore	43.6	−7	7.2	NW
Mass.	Boston	37.6	−13	11.7	W
Mich.	Alpena	29.1	−27	11.3	W
	Detroit	35.4	−24	13.1	SW
	Marquette	27.6	−27	11.4	NW
Minn.	Duluth	25.1	−41	11.1	SW
	Minneapolis	29.6	−33	11.5	NW
Miss.	Vicksburg	56.0	−1	7.6	SE
Mo.	St. Joseph	40.3	−24	9.1	NW
	St. Louis	43.3	−22	11.8	NW
	Springfield	43.0	−29	11.3	SE
Mont.	Billings	34.7	−49	....	W
	Havre	27.7	−57	8.7	SW
Neb.	Lincoln	37.0	−29	10.9	N
	North Platte	34.6	−35	9.0	W
Nev.	Tonopah	39.6	−7	9.9	SE
	Winnemucca	37.9	−28	9.5	NE
N. H.	Concord	33.4	−35	6.0	NW
N. J.	Atlantic City	41.6	−7	10.6	NW
N. Y.	Albany	35.1	−24	7.9	S
	Buffalo	34.7	−14	17.7	W
	New York	40.3	−6	13.3	NW
N. M.	Santa Fe	38.0	−13	7.3	NE

TABLE 2. CLIMATIC CONDITIONS COMPILED FROM WEATHER BUREAU RECORDS—(Continued)

COL. A	COL. B	COL. C	COL. D	COL. E	COL. F
State or Province	City	Average Temp., Oct. 1st–May 1st	Lowest Temperature Ever Reported	Average Wind Velocity Dec., Jan., Feb., Miles per Hr	Direction of Prevailing Wind, Dec., Jan., Feb.
N. C.	Raleigh	49.7	−2	7.3	SW
	Wilmington	53.1	5	8.9	SW
N. D.	Bismarck	24.5	−45	....	NW
	Devils Lake	18.9	−44	11.4	W
Ohio	Cleveland	36.9	−17	14.5	SW
	Columbus	39.9	−20	9.3	SW
Okla.	Oklahoma City	48.0	−17	12.0	N
Ore.	Baker	34.1	−20	6.0	SE
	Portland	45.9	−2	6.5	S
Pa.	Philadelphia	41.9	−6	11.0	NW
	Pittsburgh	40.8	−20	13.7	NW
R. I.	Providence	37.6	−9	14.6	NW
S. C.	Charleston	56.9	7	11.0	N
	Columbia	53.7	−2	8.0	NE
S. D.	Huron	28.1	−43	11.5	NW
	Rapid City	32.3	−34	7.5	W
Tenn.	Knoxville	47.0	−16	6.5	SW
	Memphis	50.9	−9	9.6	NW
Texas	El Paso	53.0	−2	10.5	NW
	Fort Worth	54.7	−8	11.0	NW
	San Antonio	60.7	4	8.2	N
Utah	Modena	38.1	−24	8.9	W
	Salt Lake City	40.0	−20	4.9	SE
Vt.	Burlington	29.3	−27	12.9	S
Va.	Norfolk	49.1	2	9.0	N
	Lynchburg	45.2	−7	5.2	NW
	Richmond	47.4	−3	7.4	S
Wash.	Seattle	45.3	3	9.1	SE
	Spokane	37.5	−30	....	SW
W. Va.	Elkins	38.8	−21	4.8	W
	Parkersburg	41.9	−27	6.6	S
Wis.	Green Bay	28.6	−36	12.8	SW
	La Crosse	31.2	−43	5.6	NW
	Milwaukee	33.0	−25	11.7	W
Wyo.	Sheridan	31.0	−45	5.3	NW
	Lander	28.9	−36	3.0	NE
Alta.	Edmonton	23.3	−57	4.5	W
B. C.	Victoria	43.8	−2	8.9	N
	Vancouver	41.7	2	4.2	E
Man.	Winnipeg	17.2	−46	12.4	SW
N. B.	Fredericton	27.1	−35	8.7	NW
N. S.	Yarmouth	35.5	−12	13.0	NW
Ont.	London	32.5	−26	....	....
	Ottawa	26.9	−33	7.5	W
	Pt. Arthur	21.6	−51	....	....
	Toronto	32.0	−26	13.5	SW
P. E. I.	Charlottetown	30.1	−23	8.7	NW
Que.	Montreal	27.4	−27	15.4	SW
	Quebec	24.4	−34	15.0	SW
Sask.	Prince Albert	14.7	−70	3.2	SW
Yukon	Dawson	1.6	−68	....	....

arrives. In industrial plants, quite a different condition exists, and heat sources, if they are always available during the period of human occupancy, may be substituted for a portion of the heating installation. In no case should the actual heating installation (exclusive of heat sources) be reduced below that required to maintain at least 40 F in the building.

Motors and Machinery

Motors and the machinery which they drive, if both are located in the room, convert all of the electrical energy supplied into heat, which is retained in the room if the product being manufactured is not removed until its temperature is the same as the room temperature.

If power is transmitted to the machinery from the outside, then only the heat equivalent of the brake horsepower supplied is used. In the first case the Btu supplied per hour $= \frac{\text{Motor horsepower}}{\text{Efficiency of motor}} \times 2{,}546$, and in the second case Btu per hour = bhp $\times$ 2,546, in which 2,546 is the Btu equivalent of 1 hp-hour. In high-powered mills this is the chief source of heating and is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

The heat (in Btu per hour) from electric lamps is obtained by multiplying the watts per lamp by the number of lamps and by 3.415. One cubic foot of producer gas gives off about 150 Btu per hour; one cubic foot of illuminating gas gives off about 535 Btu per hour, and one cubic foot of natural gas gives off about 1000 Btu per hour. A Welsbach burner averages 3 cu ft of gas per hour and a fish tail burner, 5 cu ft per hour. For information concerning the heat supplied by persons, see Chapter 2.

For intermittent heating allow 10 per cent additional for rooms heated in the day time only, and for unheated intervals of several days or more, add 25 per cent in determining minimum heating requirements and size of plant.

EXAMPLE OF HEAT LOSS COMPUTATIONS (See Fig. 1.)

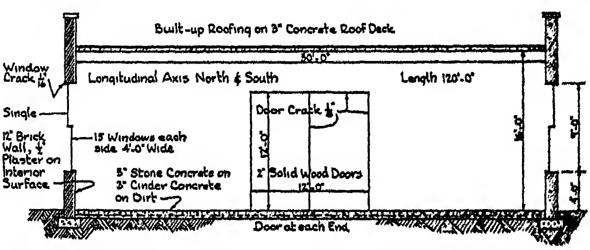

FIG. 1. ELEVATION OF FACTORY BUILDING

1. LOCATION...Philadelphia, Pa.

2. LOWEST OUTSIDE TEMPERATURE. (Table 2).......................................— 6 F

3. BASE TEMPERATURE: *In this example* a design temperature 10 deg F above lowest on record instead of 15 F is used. Hence the base temperature =

$$(-6 + 10) = +4 \text{ F}.$$

4. DIRECTION OF PREVAILING WIND (during Dec., Jan., Feb.)........................Northwest

5. BREATHING-LINE TEMPERATURE (5 ft from floor)...60 F

6. INSIDE AIR TEMPERATURE AT ROOF:

   The air temperature just below *roof* is higher than at the breathing line. Height of roof is 16 ft, or it is $16 - 5 = 11$ ft above breathing line. Allowing 2 per cent per foot above 5 ft, or $2 \times 11 = 22$ per cent, makes the temperature of the air under the roof $= 1.22 \times 60 = 73.2$ F.

7. INSIDE TEMPERATURE AT WALLS:

   The air temperature at the mean height of the *walls* is greater than at the breathing line. The mean height of the walls is 8 ft and allowing 2 per cent per foot above 5 ft, the average mean temperature of the walls is $1.06 \times 60 = 63.6$ F. By similar assumptions and calculations, the mean temperature of the *glass* will be found to be 64.2 F and that of the *doors* 61.2 F.

8. AVERAGE WIND VELOCITY (Table 2)...11.0 mph

9. OVER-ALL DIMENSIONS (See Fig. 1). ....120 x 50 x 16 ft

10. CONSTRUCTION:

    Walls—12-in. brick, with ½-in. plaster applied directly to inside surface.
    Roof—3-in. stone concrete and built-up roofing.
    Floor—5-in. stone concrete on 3-in. cinder concrete on dirt.
    Doors—One 12 ft x 12 ft wood door (2 in. thick) at each end.
    Windows—Fifteen, 9 ft x 4 ft single glass double-hung windows on each side.

11. TRANSMISSION COEFFICIENTS:

    Walls—(Table 8, Chapter 5, Wall 2B)...$U = 0.34$
    Roof—(Table 16, Chapter 5, Roofs 2A and 3A)............................$U = 0.77$
    Floor—(Table 15, Chapter 5, Floors 5A and 6A).........................$U = 0.63$
    Doors—(Table 18B, Chapter 5)..$U = 0.46$
    Windows—(Table 18A, Chapter 5)..$U = 1.13$

12. INFILTRATION COEFFICIENTS:

    Windows—Average windows, non-weatherstripped, $\frac{1}{16}$-in. crack and $\frac{3}{64}$-in. clearance. The leakage per foot of crack for an 11 mile wind velocity is 25.0 cfh. (Determined by interpolation of Table 2, Chapter 6). The heat equivalent per hour per degree per foot of crack, is from Chapter 6.

    $25.0 \times 0.018 = 0.45$ Btu per deg Fahr per foot of crack.

    Doors—Assume infiltration loss through door crack twice that of windows or $2 \times 0.45 = 0.90$ Btu per deg Fahr per foot of crack.

    Walls—As shown by Table 1, Chapter 6, a plastered wall allows so little infiltration that in this problem it may be neglected.

13. CALCULATIONS: See calculation sheet, Table 3.

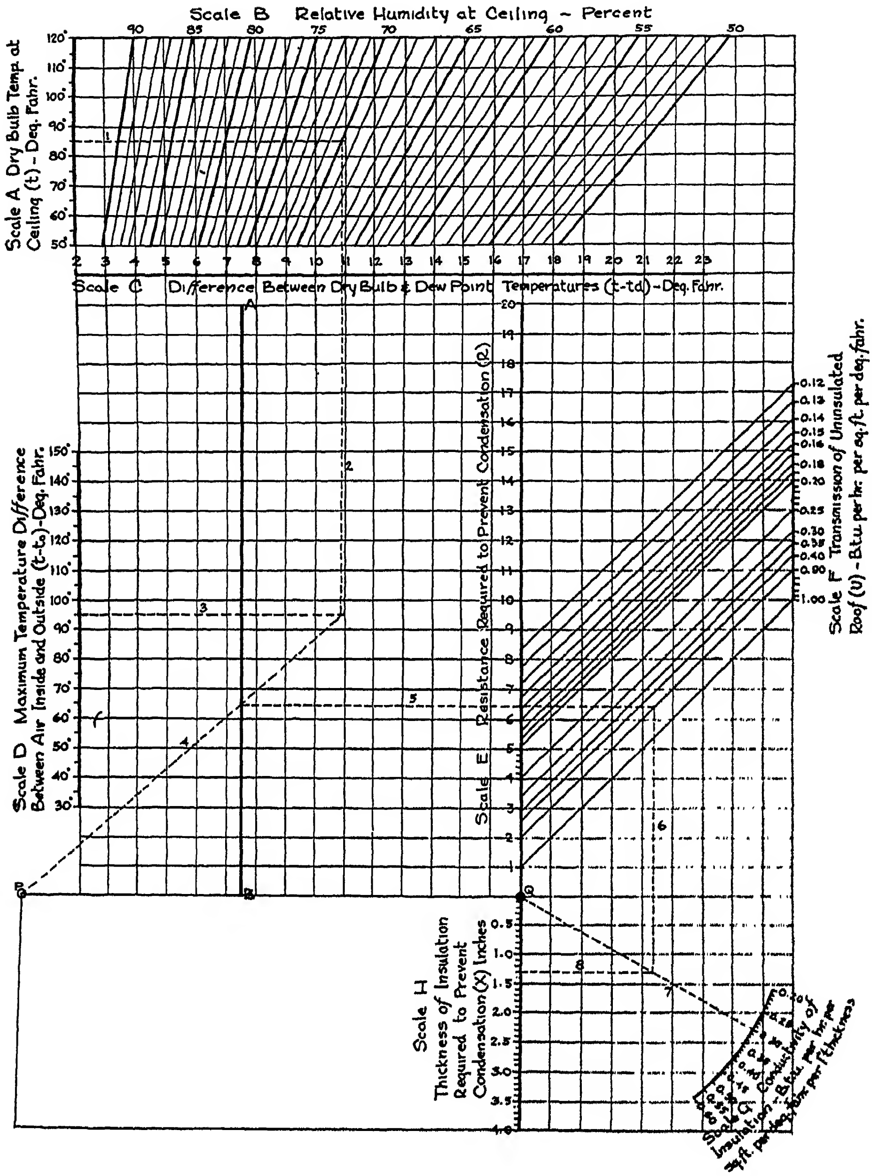

FIG. 2. CHART FOR DETERMINING THICKNESS OF INSULATION REQUIRED TO PREVENT CONDENSATION

TABLE 3. CALCULATION SHEET SHOWING METHOD OF ESTIMATING HEAT LOSSES OF BUILDING SHOWN IN FIG. 1

PART OF BUILDING	WIDTH IN FEET	HEIGHT IN FEET	NET SURFACE AREA OR CRACK LENGTH	COEFFICIENT	TEMP. DIFF.	TOTAL BTU
North Wall:						
Brick, ½ in. plaster	50	16	656	0.34	59.6	13,293
Doors (2 in. Wood)	12	12	144	0.46	57.2	3,789
⅛ in. Crack	1 pair doors		60	0.90	57.2	1,544[a]
West Wall:						
Brick, ½ in. plaster	120	16	1380	0.34	59.6	27,964
Glass (Single)	15 x 4	9	540	1.13	60.2	36,734
1/16 in. Crack	Double Hung Windows (15)		450	0.45	60.2	6,095[a]
South Wall	Same as North Wall					18,626
East Wall	Same as West Wall					70,793
Roof, 3 in. Concrete and Slag-surfaced built-up roofing	50	120	6000	0.77	69.2	319,704
Floor, 5 in. Stone Concrete on 3 in. Cinder Concrete	50	120	6000	0.63	5[b]	18,900
GRAND TOTAL of heat required for building in Btu per hour						517,442

[a]This building has no partitions and whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side. Therefore, only one-half of the total crack will be used in computing infiltration for each side and each end of building.

[b]A 5 F temperature differential is commonly assumed to exist between the air on one side of a large floor laid on the ground and the ground.

CONDENSATION ON BUILDING SURFACES[2]

Condensation on the interior surfaces of buildings is often a serious problem. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short-circuiting of electric power and lighting systems, necessitating shut-downs and incurring costly repairs. It also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected.

Condensation is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied in two ways, (1) by increasing the temperature of such surfaces above the dew-point temperature, or (2) by lowering the humidity.

Dehumidification, of course, is not permissible where a high relative humidity is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature by decreasing the inside surface resistance. This can be accomplished by increasing the velocity of air passing over the surface, or by increasing the over-all resistance of the wall or roof by installing a sufficient thickness of insulation.

The latter method is generally used, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved. This in turn is based

[2]For additional information on this subject see Preventing Condensation on Interior Building Surfaces, by Paul D. Close (A S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

on the fundamental principle that the drop in temperature is proportional to the resistance.

The chart (Fig. 2) can be used for approximating the thickness of insulation required to prevent condensation on the interior wall or roof surfaces of a building. Although this chart is intended primarily for roofs, it can be used for walls by taking the dry-bulb temperature and the corresponding relative humidity near the walls at the point which will necessitate the maximum heat resistance to prevent condensation, instead of using the temperature and humidity near the ceiling.

Example 2. Determine the thickness of insulation required to prevent ceiling condensation for the following conditions: Dry-bulb temperature near ceiling, 85 F; Relative humidity, 70 per cent; Lowest outside temperature, — 10 F; Construction of uninsulated roof, 1 in. yellow pine sheathing and built-up roofing; Coefficient of transmission of roof, 0.49; Conductivity of insulation to be used, 0.30.

Solution. The solution of this problem is indicated on the chart (Fig. 2) by the dotted line:

1. Locate the inside dry-bulb temperature of 85 F on scale *A*, and draw a line horizontally to the 70 per cent relative humidity curve, indicated on scale *B*.
2. Draw line *2* vertically downward from the intersection located in item 1.
3. Locate on scale *D* the temperature difference of 95 F between the ceiling temperature of 85 F and the lowest outside temperature of — 10 F, and draw a line horizontally until it intersects with line *2*.
4. From the point of intersection of lines *2* and *3*, draw a line to the point *P*.
5. From the intersection of lines *4* and *AB*, draw a line horizontally until it intersects with the diagonal line corresponding to a coefficient of transmission of the roof of 0.49, located on scale *F*.
6. From the intersection found by paragraph 5, draw line *6* vertically downward.
7. Locate the conductivity of 0.30 Btu per hour per square foot per degree Fahrenheit of the insulation on scale *G* and draw a line to point *Q*.
8. From the intersection of lines *6* and *7*, draw a line horizontally to scale *H*, on which the thickness of insulation of this conductivity is indicated, which is 1.3 in. The nearest commercial thickness above 1.3 in. would, of course, be selected.

Chapter 8

THE COOLING LOAD

Conditions to be Maintained, Cooling Load, Transmission with No Sun Effect, Temperatures, Sun Effect, Transmission Through Glass, Heat and Moisture Leakage, Heat and Moisture Sources

THE cooling load may be calculated in a manner similar to that used in calculating the heating load as the conditions are much the same. The direction of the flow of heat is reversed, however, and in most cases additional factors must be considered, such as the sun effect and the heat from occupants, lights, motors, and other sources. The character of the load depends on the type of building to be cooled as, for example, in auditoriums and other places of assemblage where the maximum load usually is that due to the heat and moisture given off by the occupants, or in office buildings and residences where sun effect and the transmission and infiltration of heat through the building shell are most important.

While cooling is generally identified with the summer season, it is often necessary to cool in winter as well as in summer. In a crowded place of assemblage the heat given off by the occupants, together with that given off by the lighting and power equipment, may be more than the normal heat loss through the structure even in winter under cold climatic conditions. A typical case for winter might show about 300 Btu of body heat, plus 100 Btu per person from lights, etc., being given up to the building, compared with about 200 Btu heat loss from the building, per person per hour. This would mean that 200 Btu per person must be absorbed by the air conditioning system.

Much of the basic information for the design of comfort conditioning installations has resulted from research conducted at the A.S.H.V.E. Research Laboratory and at institutions with which cooperative research investigations have been carried on. These data include the effective temperature index, and heat and moisture loss data, given in Chapter 2.

CONDITIONS TO BE MAINTAINED

The conditions to be maintained in an enclosure are variable and depend on many factors, especially the season of the year and (during the summer) the outside dry-bulb temperature and the duration of the period of occupancy. Information concerning the proper effective temperatures to be maintained for various seasons is given in Chapter 2, where are also tabulated the most desirable indoor air conditions to be maintained in summer for exposures less than three hours. (See Table 2, Chapter 2).

In installations for auditoriums and theaters the requirements are

different from those in factories, since there must be a considerable volume of air circulated in order to provide ventilation and cooling.

COOLING LOAD

The cooling load may be divided into the following parts:

1. Transmission of heat through walls, roof, glass, etc., with allowances for sun-exposed surfaces and heat capacity.
2. Transmission of solar radiation through glass and absorption by interior furnishings.
3. Heat and moisture from infiltration and from outside air introduced.
4. Heat and moisture from occupants and heat from lights, machinery and other sources.

Transmission With No Sun Effect

The transmission load *for surfaces not exposed to the sun* is calculated in a manner similar to that described in Chapter 7, by means of the following formula:

$$H_t = A\,U\,(t_o - t) \qquad (1)$$

where

H_t = heat transmitted through the material of the wall, glass, roof, or floor in Btu per hour.

A = net inside area of wall, glass, roof, or floor in square feet.

t = inside temperature, degrees Fahrenheit.

t_o = outside temperature, degrees Fahrenheit.

U = coefficient of transmission of wall, floor, roof, or glass in Btu per hour per square foot per degree Fahrenheit difference in temperature. (Tables 8 to 18, Chapter 5).

Temperatures

The maximum cooling load in summer may be calculated for an outside dry-bulb temperature not to exceed 95 F and a wet-bulb temperature not to exceed 80 F. While higher outside dry-bulb temperatures are frequently observed they are either of short duration or accompanied by low relative humidities. Weather Bureau reports are frequently misleading in this respect as they report the maximum temperature for the day with the relative humidity which occurs during a different period and which usually is much higher than the relative humidity occurring at the maximum temperature. Any statement of weather condition which gives a wet-bulb temperature higher than 80 F in the United States is questionable.

Summer dry-bulb and wet-bulb temperatures for various cities are given in Table 1. It will be noted that the wet-bulb temperatures are not the maximums but the design temperatures which should be used in air-conditioning calculations. The maximum outside wet-bulb temperatures as given in Weather Bureau reports usually occur only from 1 per cent to 4 per cent of the time, and they are therefore of such short duration that it is not practical to design a cooling system covering this range.

Sun Effect

Solar radiation is an important factor in the mechanism of heat flow into buildings. Research conducted at the A.S.H.V.E. Research Labora-

Table 1. Average Maximum Design Dry-Bulb Temperatures, Design Wet-Bulb Temperatures, Wind Velocities, and Wind Directions for June, July, August, and September

State	City	Average Maximum Design Dry-Bulb	Design Wet-Bulb	Summer Wind Velocity MPH	Prevailing Summer Wind Direction
Ala.	Birmingham	93	77	5.2	S
	Mobile	94	78	8.6	SW
Ariz.	Phoenix	110	77	6.0	W
Ark.	Little Rock	95	77	7.0	NE
Calif.	Los Angeles	88	70	6.0	SW
	San Francisco	85	68	11.0	SW
Colo.	Denver	90	64	6.8	S
Conn.	New Haven	88	74	7.3	S
D. C.	Washington	93	76	6.2	S
Fla.	Jacksonville	94	78	8.7	SW
	Tampa	94	79	7.0	E
Ga.	Atlanta	91	75	7.3	NW
	Savannah	95	79	7.8	SW
Idaho	Boise	95	65	5.8	NW
Ill.	Chicago	88	73	10.2	NE
	Peoria	91	75	8.2	S
Ind.	Indianapolis	90	73	9.0	SW
Iowa	Des Moines	92	74	6.6	SW
Ky.	Louisville	94	75	8.0	SW
La.	New Orleans	94	79	7.0	SW
Maine	Portland	85	71	7.3	S
Md.	Baltimore	93	76	6.9	SW
Mass.	Boston	88	73	9.2	SW
Mich.	Detroit	88	72	10.3	SW
Minn.	Minneapolis	84	72	8.4	SE
Miss.	Vicksburg	95	78	6.2	SW
Mo.	Kansas City	92	75	9.5	S
	St. Louis	93	76	9.4	SW
Mont.	Helena	87	63	7.3	SW
Nebr.	Lincoln	93	74	9.3	S
Nev.	Reno	93	64	7.4	W
N. J.	Trenton	93	75	10.0	SW
N. Y.	Albany	90	74	7.1	S
	Buffalo	83	72	12.2	SW
	New York	91	75	12.9	SW
N. M.	Santa Fe	87	63	6.5	SE
N. C.	Asheville	87	72	5.6	SE
	Wilmington	93	79	7.8	SW
N. D.	Bismarck	88	69	8.8	NW
Ohio	Cleveland	87	72	9.9	S
	Cincinnati	93	75	6.6	SW
Okla.	Oklahoma City	96	76	10.1	S
Ore.	Portland	83	65	6.6	NW
Pa.	Philadelphia	93	76	9.7	SW
	Pittsburgh	91	73	9.0	NW
R. I.	Providence	85	73	10.0	NW
S. C.	Charleston	94	80	9.9	SW
	Greenville	93	76	6.8	NE
Tenn.	Chattanooga	94	76	6.5	SW
	Memphis	93	77	7.5	SW

TABLE 1. AVERAGE MAXIMUM DESIGN DRY-BULB TEMPERATURES, DESIGN WET-BULB TEMPERATURES, WIND VELOCITIES, AND WIND DIRECTIONS FOR JUNE, JULY, AUGUST, AND SEPTEMBER (Continued)

STATE	CITY	AVERAGE MAXIMUM DESIGN DRY-BULB	DESIGN WET-BULB	SUMMER WIND VELOCITY MPH	PREVAILING SUMMER WIND DIRECTION
Texas	Dallas	99	76	9.4	S
	Galveston	93	79	9.7	S
	San Antonio	100	78	7.4	SE
	Houston	93	79	7.7	S
	El Paso	98	69	6.9	E
Utah	Salt Lake City	95	67	8.2	SE
Vt.	Burlington	85	71	8.9	S
Va.	Norfolk	91	76	10.9	S
	Richmond	95	76	6.2	SW
Wash.	Seattle	83	61	7.9	S
	Spokane	89	63	6.5	SW
W. Va.	Parkersburg	90	74	5.3	SE
Wis.	Madison	89	73	8.1	SW
	Milwaukee	87	72	10.4	S
Wyo.	Cheyenne	85	62	9.2	S

tory[1] has shown that a large error may be introduced into the calculations by failure to consider the periodical character of heat flow resulting from the diurnal movement of the sun and the heat capacity of the structure, which determine the timing and magnitude of the heat wave flowing through the wall into a building on a hot, sunny day.

Unfortunately, despite intensive study, all data on solar radiation

TABLE 2. HEAT EQUIVALENTS OF VARIOUS DEVICES, BTU

Lights and electric appliances	3415 per kilowatt
Motors	2546 per horsepower hour
Restaurant coffee urns, 10-gal capacity	16000 per hour
Dish warmers per 10 sq ft of shelf	6000 per hour
Restaurant range—4 burners and oven	100,000 per hour
Residence gas range	
Giant burner	12000 per hour
Medium burner	9000 per hour
Oven	1000 per cu ft of space
Pilot	250 per hour
Electric Range	
Small burner	3412 per hour
Medium burner	4100 per hour
Large burner	7700 per hour
Oven	10236 per hour
Appliance connection	2250 per hour
Warming compartment	1023 per hour

[1]For further information on this subject see following A.S.H.V.E. research papers: Coefficients of Heat Transfer as Measured under Natural Weather Conditions, by F. C. Houghten and C. G. F. Zobel (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928); Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930); Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by F. C. Houghten, J. L. Blackshaw, E. M. Pugh and Paul McDermott (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

through building walls are too theoretical and mathematically complicated to be of much practical value to the heating and ventilating engineer. Accordingly, the customary rule-of-thumb method of adding an arbitrary 25 F to the dry-bulb temperature difference in calculating the heat transmission through a wall or roof which may be exposed to the sun for any appreciable length of time is given as a workable solution to an exceedingly complex problem.

Solar Radiation

Fig. 1 shows the total amount of solar energy in Btu per square foot per hour received during the day by a surface normal to the rays of the sun, by a horizontal surface, and by east, west, and south walls. The curves are drawn from A.S.H.V.E. Laboratory data obtained by pyrheliometer, are based on sun time, and are for a perfectly clear day on August 1 at a north latitude of 40 deg. Data from these curves may be used with little error for most United States latitudes and for all of the hotter months of the year.

The absorption of solar radiation by an interior surface depends upon the character of the surface, the angle of the surface with respect to the direction of the radiation, and the angle and type of glass through which the radiant rays pass. The heat absorption by a black oilcloth surface perpendicular to the sun's rays was found to be as high as 273 Btu per square foot per hour, based on tests conducted by the A.S.H.V.E. Research Laboratory in Pittsburgh[2]. Lamp black, red brick dust, and aluminum bronze painted surfaces perpendicular to the sun's rays showed respectively 94.0, 63.4, and 28.2 per cent as high a rate of absorption as the black oilcloth.

Transmission Through Glass

In considering the effect of glass on heat absorption, several factors must be considered. As the sun's rays impinge against and pass through an intervening sheet of glass, some radiation is reflected directly from each of the two glass surfaces, and some is absorbed by the glass depending upon its character and thickness. All of the heat reflected by the glass surfaces and most of that absorbed within the glass is stopped from passing through the glass into a room.

The A.S.H.V.E. tests indicated that a single pane of double strength glass absorbs from 8.9 to 16.5 per cent of the solar radiation passing through it when the impingement is normal. For smaller angles of impingement, the glass retards percentages of the total radiant energy approximately in proportion to the sine of the angle. Experiments[3] indicate a glass absorption of 16.7 per cent for one pane of glass and 37.5 per cent for two ¼-in. panes separated by a 1¾-in. air space.

In a recent paper by the A.S.H.V.E. Research Laboratory[4] it was shown

[2]Absorption of Solar Radiation in Relation to the Temperature, Color, Angle, and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

[3]Field Studies of Office Building Cooling (A.S.H.V.E. Research Paper), by J. H. Walker, S. S. Sanford, and E. P. Wells (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

[4]Radiation of Energy Through Glass by J. L. Blackshaw and F. C. Houghten (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, October, 1933).

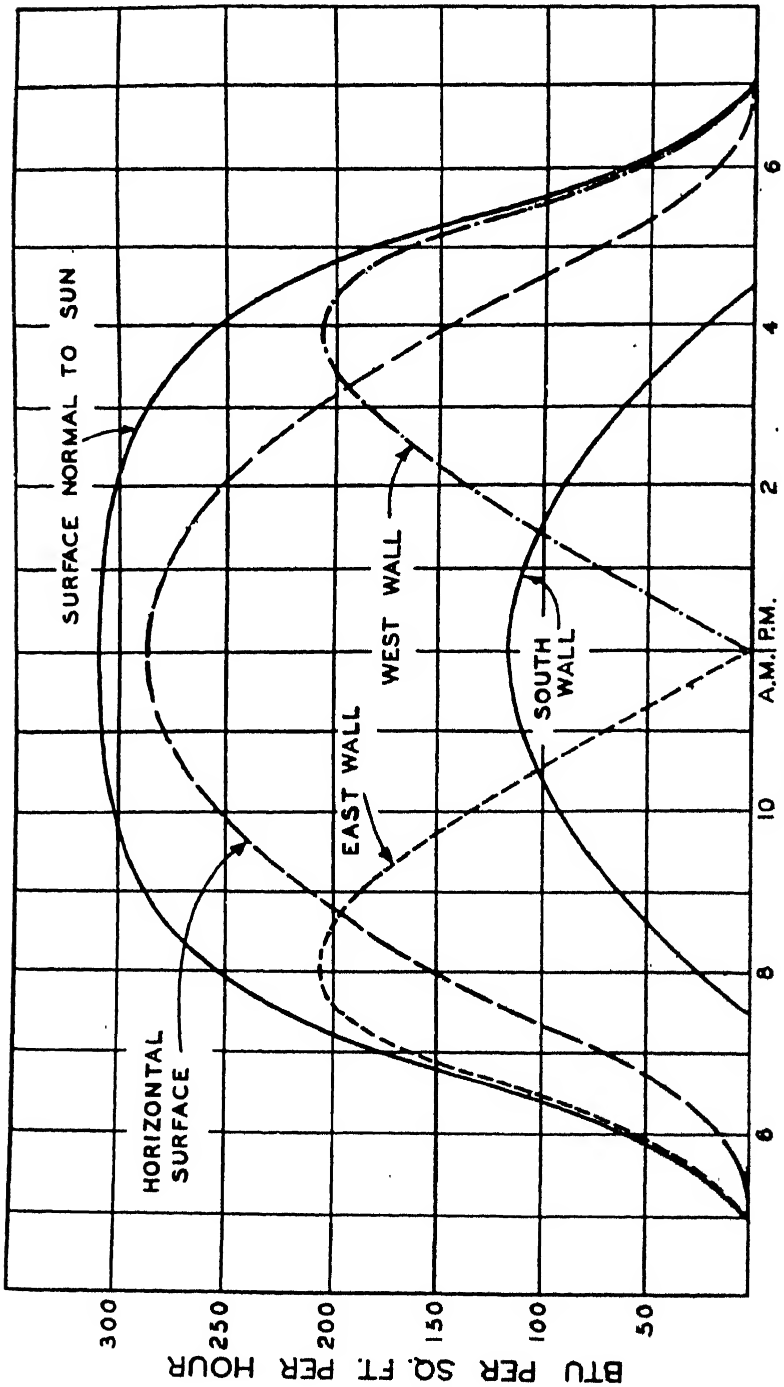

Fig. 1. Curves Giving Solar Intensity Normal to Sun, on Horizontal Surface and on Walls for August 1

that ordinary double strength window glass transmits no measurable amount of energy radiated from a source at 500 F or lower; that it transmits only 6.0 and 12.3 per cent of the total radiation from surfaces at 700 F and 1000 F, respectively; and that it transmits 65.7 per cent of the radiation from an arc lamp, 76.3 per cent of the radiation from an incandescent tungsten lamp, and 89.9 per cent of the radiation from the sun. Thus, glass windows in a room constitute heat traps, which allow rather free transmission of radiant energy into the room from the sun to warm objects in it, but do not allow the transmission of re-radiated heat from these same objects.

Some recent tests[3] indicated that sunshine through window glass is the most important factor to contend with in the cooling of an office building. At times it was shown to account for as much as 75 per cent of the total cooling necessary. Because of the importance of the sunshine load, cooling systems should be zoned so that the side of the building on which the sun is shining can be controlled separately from the other sides of the building. If buildings are provided with awnings so that the window glass is shielded from sunshine, the amount of cooling required will be reduced and there will also be less difference in the cooling requirements of different sides of the building.

Where the glass area is not great, the rule-of-thumb method of adding an arbitrary 25 F to the design dry-bulb temperature difference may be applied. The heat flow may then be figured as if there were no sun effect.

Heat and Moisture Leakage

An allowance must be made for the heat and moisture in the outside air introduced for ventilating purposes or entering the building through cracks, crevices, doors, and other places where infiltration might occur. The volume of air entering due to infiltration may be estimated from data given in Chapter 6, and information on the amount of outside air required for ventilation will be found in Chapter 2.

The heat gain resulting from the outside air introduced may be estimated from the following formula:

$$H_i = Qd_o\,(\Theta_o - \Theta) \qquad (2)$$

where

H_i = heat to be removed from outside air entering the building, Btu per hour.
Q = volume of outside air entering the building, cubic feet.
d_o = density of outside air, pounds per cubic foot, at the temperature t_o.
Θ_o = heat content of mixture of outside dry air (at temperature t_o) and water vapor, Btu per pound of dry air.
Θ = heat content of mixture of inside dry air (at temperature t) and water vapor, Btu per pound of dry air.

Heat and Moisture Sources

Figs. 4 to 7, Chapter 2, show the heat and moisture given off by human beings under various conditions of activity. For average conditions where

[3]Footnote p. 119.

a person is normally at rest, as in a theater, or doing very light work, as in a restaurant or residence, the heat given off will vary from 700 to 1200 grains of moisture per hour. Examples illustrating heat and moisture loss calculations for human beings are given in Chapter 2.

All sources of heat must of course be considered in designing the conditioning system. The heat equivalents of various devices are given in Table 2.

An example of the calculation of the cooling load, is given in Chapter 9.

Chapter 9

CENTRAL FAN AIR CONDITIONING SYSTEMS

Types of Systems, Dehumidifier, By-Pass, Local Recirculation, Surface Cooling Unit Method

CENTRAL fan systems, equipped for cooling and dehumidifying, are used principally in the air conditioning of theaters, restaurants, office buildings, or other places where many people gather, and in manufacturing establishments where air conditions have an important influence on the quality of the product. The design of such systems is considered in this chapter, while in Chapter 22 central fan systems for heating and humidifying are described.

TYPES OF SYSTEMS

The four most common types of central fan cooling and dehumidifying systems are as follows:

1. All air through dehumidifier or air washer.
2. By-pass method, by which part of the air is put through the dehumidifier and mixed with air from other sources.
3. Local recirculation.
4. Surface cooling.

There are, of course, many other methods which may be employed such as dehumidifying air by passing it through or over a dehydrating agent, or passing air directly over ice, or by evaporative cooling. (See Chapters 10 and 11).

DEHUMIDIFIER METHOD

The dehumidifier system is simple, comprising a dehumidifier, fan, reheaters and necessary accessories, such as pumps, motors, piping and controls. The return air from the conditioned enclosure is mixed with the outside air and carried through the dehumidifier and cooled. It may then be reheated by various methods to the predetermined delivery conditions. Information concerning dehumidifiers will be found in Chapter 11.

Example 1. Dehumidifier Method. (See Figs. 1 and 2). The restaurant shown in Fig. 1 has a seating capacity of 300 people and there are 20 employees. The lighting load is 4200 watts and the lights are used continuously during the day.

The service counter is equipped with an exhaust system with a capacity of 1500 cfm which is used to remove all the heat and moisture produced by coffee urns, steam tables, plate warmers, toasters, and grilles.

The windows along the north and west streets are colored glass and used only for decorative purposes. Each window is 5 ft wide by 2 ft deep. The show windows on the

south street and the entrance windows and the first windows on the west street are 9 ft high.

The overall heat transmission factors, U, for the different walls of the restaurant are determined by the methods outlined in Chapter 5 and are found to be the following: Exterior walls, 0.221; interior wall, 0.296; floor, 0.433; and ceiling, 0.299.

Determine the dehumidification, cooling and reheating loads, considering the outside dry-bulb temperature to be 95 F and the outside design wet-bulb temperature to be 75 F.

Solution. According to Table 2, Chapter 2, the inside dry-bulb temperature should be 80 F and the wet-bulb temperature should be 65 F.

The heat gain to be considered may be calculated as follows:

BUILDING

South Wall

Total area = 27 × 15 = 405 sq ft
Glass (including entrance windows)
2 (9 × 9) + 2 (5 × 9) + (door) (6 × 9) = 306 sq ft
Net wall area = 405 − 306 = 99 sq ft
Transmission factor for wall = 0.221
Transmission factor for glass = 1.13
Temperature difference = 15 + 25 (for sun effect) = 40 deg

	Heat Gain, Btu per Hour
99 × 0.221 × 40 = 875 Btu per hour	
306 × 1.13 × 40 = 13831 Btu per hour	14706

West Wall

Total area = 100 × 15 = 1500 sq ft
Glass 15 × (2 × 5) + (9 × 5) = 195 sq ft
Net wall area = 1500 − 195 = 1305 sq ft

1305 × 0.221 × 40 = 11536 Btu per hour	
195 × 1.13 × 40 = 8814 Btu per hour	20350

North Wall (no sun effect)

Total wall area = 27 × 15 = 405 sq ft
Glass 2 × (2 × 5) = 20 sq ft
Net wall area = 385 sq ft

385 × 0.221 × 15 = 1276 Btu per hour	
20 × 1.13 × 15 = 339 Btu per hour	1615

East Wall (interior, no sun effect)

Total wall = 100 × 15 = 1500 sq ft

1500 × 0.296 × 15 =	6660

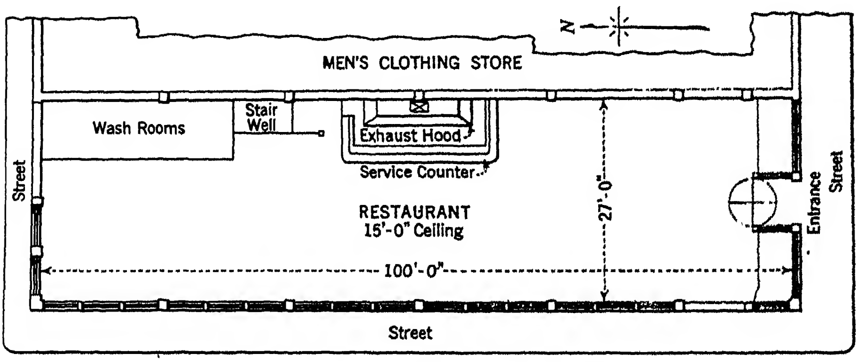

FIG. 1. FLOOR PLAN OF RESTAURANT

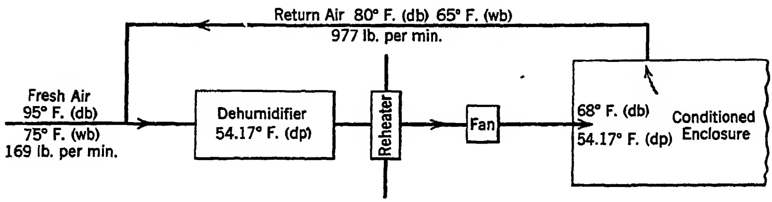

FIG. 2. DIAGRAM OF DEHUMIDIFIER METHOD

Floor

100 × 27 = 2700 sq ft
2700 × 0.433 × 15 = 17537

Ceiling

100 × 27 = 2700 sq ft
2700 × 0.299 × 15 = 12110

INFILTRATION

The windows are sealed in metal frames set in concrete and the infiltration may be considered negligible, particularly as the prevailing wind is at a minimum when the temperature is high. The revolving entrance door, however, does present an infiltration problem.

The door is 8 ft high and 6 ft wide and during rush hours can be assumed to revolve approximately one turn per person entering. This figure is purely an assumption but the percentage error would not be more than 2 or 3 per cent of the total heat load.

The volume of the door $= \frac{\pi (6^2)}{4} \times 8 = 226$ cu ft

With 300 persons per hour entering this would be equivalent to $\frac{226 \times 300}{60} = 1130$ cfm.

A volume of 1130 cfm of air cooled from a wet-bulb temperature of 75 F to a wet-bulb temperature of 65 F would be equivalent to the removal of heat as follows:

Total heat at 75 F wet-bulb = 37.81 Btu per hour. (See Table 2, Chapter 1).
Total heat at 65 F wet-bulb = 29.62 Btu per hour

8.19 Btu per hour to be removed

$\frac{1130}{14.3}$ cfm × 8.19 × 60 = 38,831 Btu per hour

Half of this, however, would be provided for by the outside air taken into the cooling unit so the heat gain to be considered under infiltration would be 19,415 Btu per hour.

Lights

4200 × 3.415 = 14,343 Btu per hour

PEOPLE

Fig. 4, Chapter 2, shows the heat loss from the human body at different effective temperatures. For an effective temperature of 73.4 deg for persons at rest the heat loss is 400 Btu per hour per person of which 225 Btu per hour is sensible and 175 Btu per hour is latent heat. As there are 300 occupants and 20 employees the heat gain would be

(300 + 20) × 225 = 72,000 Btu per hour of sensible heat
(300 + 20) × 175 = 56,000 Btu per hour of latent heat

Fig. 6, Chapter 2, shows the moisture loss from human bodies and at 80 F to be about 1200 grains per hour per person at rest. The total moisture gain would then be (300 + 20) × 1200 = 384,000 grains per hour.

cool air to give the proper delivery temperature. By cooling a portion of the total amount of the air, the dehumidifier, pump, motors and water piping are smaller than in the system using all the air through the washer. As the major portion of the reheating is done by the by-passed air, smaller size reheaters are used. This type of system provides a constant supply of air regardless of load, and a decrease in the total refrigeration required for the season.

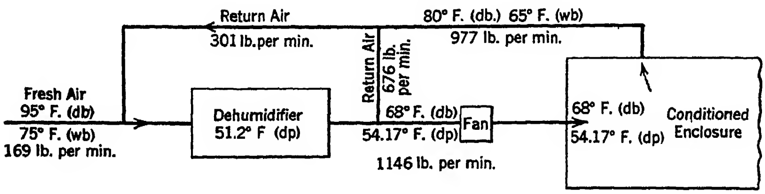

FIG. 3. DIAGRAM OF BY-PASS METHOD

Example 2. By-pass Method. (Fig. 3). Same data as Example 1. Instead of passing all the air through the dehumidifier to be cooled, a portion of it is passed through and the balance is mixed with the conditioned air at the leaving end of the dehumidifier, the mixture being so proportioned that the resultant conditions will be those required to give proper maintained conditions in the enclosure.

Setting forth the conditions as shown in Fig. 3, it is now necessary to calculate the quantity of air to be passed through the dehumidifier, the quantity by-passed, and the dew-point temperature it is necessary to carry in the dehumidifier to give the conditions sought.

There are three unknown quantities to be determined and these may be solved in two successive steps.

Let

x = the percentage of air to be by-passed.
y = the percentage of total air through the dehumidifier.
t = the dew-point temperature to be maintained in the dehumidifier.

The quantity x of 80 F air must mix with y quantity of dehumidified air to give a resultant of 65 F. Also, x quantity of air at $56\frac{1}{2}$ F (dp) must be mixed with y quantity of dehumidified air to give a resultant of 54.17 F (dp). It is assumed, of course, that the air passing through the dehumidifier is saturated, that is, the dry-bulb, wet-bulb and dew-point temperatures are the same.

Therefore,

$$80x + yt = 68 \tag{1a}$$

$$56.5x + yt = 54.17 \tag{1b}$$

$$23.5x + 0 = 13.83 \tag{1c}$$

$$x = \frac{13.83}{23.5} = 59 \text{ per cent of air by-passed.}$$

$$y = (1 - x) = 41 \text{ per cent of total air through washer.}$$

The second step is to determine the dew-point temperature in the dehumidifier. Substituting in either Equation 1a or 1b and solving will give the desired results as follows:

$$(80 \times 0.59) + t \times 0.41 = 68 \tag{2a}$$

$$t = \frac{68 - 47}{0.41} = 51.2 \text{ F (dp)} \tag{2b}$$

As the total weight of air is 1146 lb per minute and 41 per cent is required through the dehumidifier, the work done would be

$$
\begin{aligned}
1146 \times 0.41 &= 470 \text{ lb through dehumidifier} \\
&- 169 \text{ lb fresh air} \\
\hline
&301 \text{ lb recirculated air}
\end{aligned}
$$

Refrigeration required would be

$$
\begin{aligned}
\text{Total heat at 65 F} &= 29.85 \\
\text{Total heat at 51.2 F} &= 20.85 \\
\hline
&\ 9.00 \text{ Btu per pound}
\end{aligned}
$$

$$301 \times 9.0 = 2709 \text{ Btu per minute}$$

$$
\begin{aligned}
\text{Total heat at 75 F} &= 37.72 \text{ Btu per pound} \\
\text{Total heat at 51.2 F} &= 20.85 \text{ Btu per pound} \\
\hline
&\ 16.87 \text{ Btu per pound}
\end{aligned}
$$

$$
\begin{aligned}
169 \times 16.87 &= 2851 \text{ Btu per minute} \\
2709 + 2851 &= 5560 \text{ Btu per minute, total}
\end{aligned}
$$

$\frac{5560}{200} = 27.80$ tons of refrigeration required. This is 63 per cent of that required when all the air passed through the washer.

LOCAL RECIRCULATION METHOD

The local recirculation system cools and dehumidifies a small portion of air and by means of nozzles which are usually specially designed, the air is introduced into the conditioned enclosure at a high velocity (1200 to 1500 fpm) in such a way as to induce the air in the enclosure to circulate and mix with it to give the proper predetermined conditions.

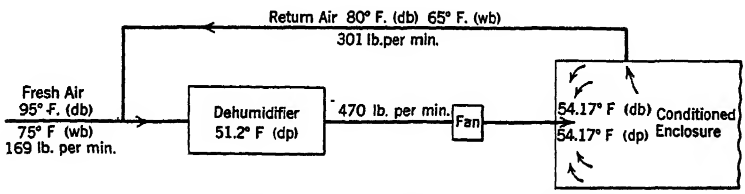

FIG. 4. DIAGRAM OF LOCAL RECIRCULATION METHOD

Example 3. Local Recirculation Method. (Fig. 4). Same data as Example 1. With this method a small quantity of air is used and discharged into the enclosure at a relatively high velocity to give the air in the room an induced circulation. This is done by omitting the reheaters and discharging the air into the enclosure at the dew-point temperature.

With the cut-and-try method it is found that a dew-point temperature of 51.2 F is necessary to pick up the heat and moisture in the enclosure, that is. the dry-bulb and dew-point temperatures are the same.

$$\frac{199736 \times 55.2}{(80 - 51.2) \times 60 \times 13.35} = 470 \text{ lb per minute}$$

The 470 lb per minute is the same as that required through the dehumidifier in the by-pass system and as the temperature is the same, the refrigeration load will be the same, the only difference between the two methods being the additional fan horsepower in the by-pass method, which is negligible, and the larger sized duct work.

SURFACE COOLING UNIT METHOD

The fourth type of system employing a surface cooling unit of the extended-surface type will cool and dehumidify. A refrigerant is circulated or expanded within the unit and air is blown over it. This system has the advantages of lower initial cost and low operating cost, and it does not require as much attention as a spray-type system.

For comfort cooling, water is usually used as the refrigerant. If a refrigerant with a temperature lower than 32 F is used, care must be exercised in the design to prevent frosting. Low temperature refrigerants often reduce the moisture content of the air lower than is usually required. Water at 45 to 50 F is quite practical, using a 10 to 15 deg rise in water temperature depending on the quantity of water used and the velocity of the water required through the tubes.

Although surface cooling units have advantages over the conventional spray-type systems, they are usually adaptable for both cooling and heating, and they can be used to control humidity in summer. The effective cooling accomplished by the unit is dependent upon many variable factors. The air velocity through the unit, air temperature, moisture content, water temperature, and velocity of the water through the tubes must all be considered in designing the unit. If any of these factors vary without a corresponding variation of the other factors, the effective cooling obtained by the unit will drop off.

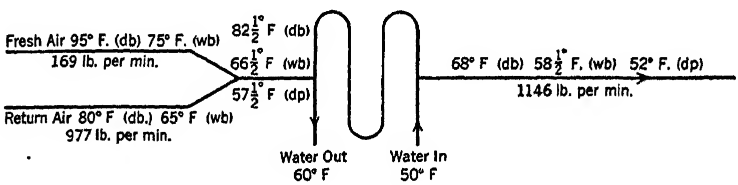

FIG. 5. DIAGRAM OF SURFACE COOLING METHOD

Example 4. Surface Cooling Method. (Fig. 5). Assuming the same required conditions and load as in Example 1, a surface cooling unit of proper size and capacity may be determined by the following calculations:

Outside air........................ 169 lb per minute at 95 F (db), 75 F (wb)
Recirculated air.............. 977 lb per minute at 80 F (db), 65 F (wb)
Air leaving unit...............1146 lb per minute at 68 F (db), 59.56 F (wb), 54.17 F (dp)
Water, in..............................50 F
Water, out.........................60 F

Water flowing counter to air flow.

It will be noticed from Fig. 5 that the dew-point temperature of the entering mixture is lower than the leaving water temperature but higher than the entering water temperature. Condensation of the moisture in the entering air will therefore occur some place in the unit where the temperature of the pipe or surface is lower than 57½ F. Condensation will continue to the leaving end of the unit and will result in a dew-point temperature of the leaving air between 50 F and 54.17 F, or approximately 52 F. With this dew-point temperature and the dry-bulb temperature of the air leaving of 68 F, which has been set by pre-determined calculations, the new wet-bulb temperature will be 58½ F*.

*It can readily be shown that if the entering water temperature were 54.17 F and the other conditions were such that the leaving dew-point temperature were 54.17 F as desired, the refrigeration load would be the same as for the by-pass and local recirculation systems. These conditions, however, have not been approached in practice to date.

The total heat to be removed by the water will be

$$\begin{aligned} \text{Total heat at } 66\tfrac{1}{2}\text{ F (wb)} &= 30.77 \\ \text{Total heat at } 58\tfrac{1}{2}\text{ F (wb)} &= \underline{25.20} \\ & \quad 5.57 \text{ Btu per pound of air} \end{aligned}$$

$$1146 \times 5.57 = 6383 \text{ Btu per minute}$$

$$\text{Refrigeration} = \frac{6380}{200} = 31.90 \text{ tons}$$

$$\text{Quantity of water required} = \frac{6380 \text{ Btu per minute}}{10 \text{ deg} \times 8.33} = 76.6 \text{ gpm}$$

As the cooling efficiency is dependent partly on the velocity of water through the tubes, this must be determined from a manufacturer's catalog. For instance, one manufacturer gives the formula

$$V = \frac{(\text{gallons per minute}) \times 1.235}{36} \qquad (3)$$

where

V = velocity in feet per second.
36 = number of tubes per unit.
1.235 = a constant for the particular sized pipe used.

Using Formula 3, the velocity would be,

$$V = \frac{76.6 \times 1.235}{36} = 2.63 \text{ fps}$$

To determine the amount of cooling surface required it is necessary to ascertain the mean temperature difference between the air and water, and the coefficient of transmission. The formula for the mean temperature difference is:

$$T_d = \frac{(T_1 - T_2) - (T_3 - T_4)}{\text{Log}_e\left(\dfrac{T_1 - T_2}{T_3 - T_4}\right)} \qquad (4)$$

where

T_d = mean temperature difference.
T_1 = entering dry-bulb temperature of air.
T_2 = leaving water temperature.
T_3 = leaving dry-bulb temperature of air.
T_4 = entering water temperature.

$$T_d = \frac{(82\tfrac{1}{2} - 60) - (68 - 50)}{\text{Log}_e\left(\dfrac{82\tfrac{1}{2} - 60}{68 - 50}\right)} = \frac{22\tfrac{1}{2} - 18}{\text{Log}_e\left(\dfrac{22\tfrac{1}{2}}{18}\right)} = 20.2 \text{ F}$$

The coefficient of transmission is usually taken from the manufacturers data, knowing the water velocity through the unit and the air velocity over the tubes. The velocity of the water through the tubes has been determined, and assuming a velocity of 600 fpm for the air the coefficient of transmission is (from manufacturer's curves):

K = 9.7 Btu per square foot surface per mean temperature difference.

The cooling surface required is usually based upon the sensible heat load. The latent heat due to condensation is taken out at the same time the sensible heat is extracted, and no extra surface is required unless the latent heat exceeds approximately 40 per cent of the total heat. This is due to the higher coefficient of transmission factor because of the wetted surface. This factor holds fairly consistent up to a point where the latent heat

approaches 40 per cent of the total heat load to be removed. If this amount is exceeded. approximately 10 per cent surface should be added.

$$\text{Surface required } (S) = \frac{H_s}{KT_d} \tag{5}$$

where

S = surface area, square feet.
H_s = sensible heat load, Btu.
K = coefficient of transmission.

The sensible heat load may be determined by subtracting the latent heat from the total heat.

Total heat removed = 6383 × 60 = 382,980 Btu per hour.

Latent heat at 57½ F (dp) = 10.70 (From Table 2, Chapter 1).
Latent heat at 52 F (dp) = 8.75 (From Table 2, Chapter 1).
1.95 Btu per pound

1.95 × 1146 × 60 = 134,082

Subtracting this from the total heat = 382,980
− 134,082

Sensible heat 248,898 Btu per hour

$$S = \frac{248{,}898}{9.7 \times 20.2} = 1270 \text{ sq ft of surface.}$$

As the latent heat load is 35 per cent of the total heat load, it is not necessary to add 10 per cent to this surface.

For information on the control of air conditioning systems, see Chapter 14.

Chapter 10

COOLING METHODS

Methods of Cooling Air, Evaporative Cooling, Dehumidification by Adsorption and Absorption, Silica Gel Systems, Alumina Systems, Refrigeration, Air Conditioning Applications, Refrigeration Machines, Evaporators, Compressors, Condensers, Amount of Cooling Water, Refrigerants, Methods of Cooling, Air Coolers, Water Coolers, Indirect Coolers, Steam Jet System, Ice for Air Cooling

FROM a study of the data in Chapter 2 and in Chapter 8 it is apparent that a reduction of effective temperature may be produced by any one of the following methods or combinations thereof:

a. Lowering of the dry-bulb temperature by the removal of sensible heat without change of the dew-point temperature (sensible cooling).

b. Lowering the dew-point temperature by the removal of moisture without change of the dry-bulb temperature (dehumidifying).

c. Lowering of the dry-bulb temperature through the evaporation of moisture without the addition or the subtraction of heat (evaporative cooling).

d. Increasing the air motion over the body with the consequenting higher rate of evaporation from the skin (air motion).

As an example let the condition be considered of 92 F dry-bulb, with a 40 per cent relative humidity, corresponding to a wet-bulb temperature of 72.8 F, and an effective temperature for still air of 81.1 F. This *effective temperature* may be reduced 3.1 F by any of the four basic methods mentioned as follows:

First, by lowering the dry-bulb temperature to 85.5 F without changing the dew-point of 64.2 which gives an effective temperature of 78 F.

Second, by reducing the moisture content of the air to 46 grains per pound of dry air without changing the dry-bulb temperature which gives an effective temperature of 78 F.

Third, by reducing the dry-bulb temperature to 83.8 F without changing the total heat of the air, requiring the evaporation of 14 grains of moisture per pound of dry air, when the effective temperature again will be 78 F.

Fourth, by increasing the air movement from still air to 460 fpm, velocity which will reduce the effective temperature 3.1 F from 81.1 F to 78 F.

Best Method to Employ

The best method of reducing the effective temperature in any specific case will always depend on the accompanying circumstances and only can be determined by the thorough analysis of a competent engineer. Generally speaking, the removal from the air of the sensible heat, or moisture, or both, by sensible cooling or dehumidifying is the most satisfactory method. Adequate results by the utilization of air motion, or by evaporative cooling, are difficult to obtain because of the dependence

of both methods upon climatic conditions beyond the engineers control although these methods are much less expensive than the first two mentioned. Cooling by evaporation is satisfactory only when the air to be cooled is very dry; air motion as a means of producing cooling effect is never entirely adequate in the range of high temperatures. Of the two, evaporative cooling, or adiabatic saturation of the air, is a much more dependable method and will reduce the effective temperature much more than will an increasing air motion within permissible limits.

As an example of this, consider an outdoor condition of 96 F dry-bulb and 80 F wet-bulb. The effective temperature under this condition is 85.7 F and, if the still air is moved with a velocity of 300 fpm, the effective temperature will be reduced only 2.0 F while saturation at the wet-bulb temperature reduces the effective temperature 5.7 F. At 300 fpm velocity this saturated air will reduce the effective temperature to 75.6 F, or a total improvement of 10.1 F.

Evaporative Cooling

Evaporative cooling is accomplished by passing air through a water spray, the water being continually re-circulated. The air entering in an unsaturated condition, evaporates a part of the water at the expense of the sensible heat. As this is an adiabatic transfer, the total heat content of the air remains constant, while the dew-point rises and the dry-bulb falls until the air is saturated. A system[1] of ducts and a propelling fan are used to distribute the air in a proper manner.

It will be seen that the reduction in dry-bulb temperature is a direct function of the wet-bulb depression of the air entering the dehumidifier and that the resulting air temperature is governed entirely by the entering wet-bulb temperature of the outside air. Often it is possible to reduce the dry-bulb temperature by as much as 18 to 20 F and just as often impossible to reduce the temperature more than 2 to 3 F.

Dehumidification by Adsorption and Absorption

Dehumidification may be accomplished in three ways,

1. By cooling the air below the dew-point and causing a part of the moisture contained to precipitate.
2. By extracting the moisture entirely, or in part, by absorption.
3. By extracting the moisture entirely, or in part, by adsorption.

As used in this discussion, the term *adsorption* pertains to the action of a substance in condensing a gas or vapor and holding the condensate on its surfaces without any change in the chemical or physical structure of the substance and with the release of sensible heat. The term, *absorption*, implies a change in the chemical or physical structure in the process of dehydration. Adsorbers include silica gel, lamisilite, etc.; absorbers include sulphuric acid.

Silica Gel System of Adsorption

Silica gel is a chemical composition made from sodium silicate and acid,

[1]See Air Washer Performance in Chapter 11; also Theory of Atmospheric Cooling in same chapter.

the chemical formula being SiO_2 and has an appearance greatly resembling that of clear quartz sand but differing in structure in that the crystals are highly porous the voids constituting 41 per cent by volume although the pores are microscopic in size. This material possesses the property of being able to adsorb a substantial portion (about 25 per cent) of its own weight of moisture drawn from the air without any increase in the silica gel volume. After the silica gel has become "saturated" or has adsorbed moisture to the limit of its capacity, the moisture may be driven off by the application of heat, again without change in the structure, volume or chemical composition of the adsorbing medium. Thus the cycle may be repeated indefinitely and when applied to air conditioning the medium is exposed to the air which results in reducing the moisture content in the

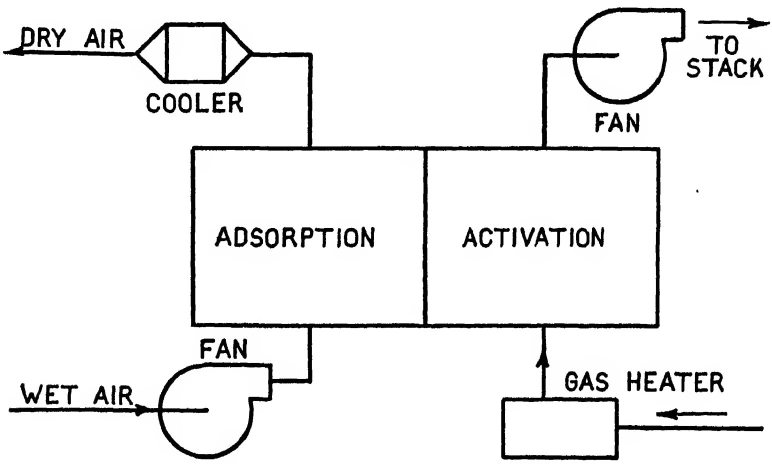

FIG. 1. SILICA GEL AIR-CONDITIONING SYSTEM—SINGLE STAGE ADSORPTION

air and releasing sensible heat which may be readily removed, thus reducing the wet-bulb or total heat of the air. A typical diagram is shown in Fig. 1.

Practical Application of Silica Gel

Silica gel has been used to replace refrigeration in one of two applications. In the one principally used, the air, from which moisture is to be extracted, is taken through silica gel beds by means of suction or pressure fans and by means of this process, the moisture becomes adsorbed by the silica gel and the air leaves at a lower dew-point and a higher sensible temperature. If this air is passed over surface coolers in which tap water or another cooling medium is flowing through the tubes, a certain amount of sensible heat will be removed. The air leaves the surface cooler or interchanger with the same dew-point with which it emerged from the silica gel beds, but with a lower dry-bulb temperature, altho the dry-bulb temperature may be higher than the temperature of the air entering the silica gel beds.

In another method, the first two of the steps outlined previously are duplicated and in addition, the air is carried through a spray type washer.

As the air enters the washer with a low wet-bulb, and as adiabatic saturation will take place at a temperature close to the entering wet-bulb, considerable cooling of the air can be accomplished but with a consequent increase of the dew-point.

It is necessary to reactivate the silica gel after it has absorbed about 25 per cent of its own weight in the form of moisture. As reactivation requires a high temperature and since silica gel is only active at low temperatures, cooling of the beds must also be completed before they can be used again. This necessitates three stages in the silica gel containers and requires either three beds of silica gel or one bed divided and automatically put in position. The reactivation is usually done by means of gas or oil fires and the cooling of the beds by means of indirect water cooling or by means of small quantities of dehydrated air taken from the system beyond the interchanger.

Alumina System of Adsorption

Activated alumina contains a trifle over 91 per cent of aluminum oxide, $Al_2 O_2$ and this material will absorb nearly 100 per cent of the vapor in air up to about 8 to 10 per cent of the weight of the adsorbing material after which the adsorption falls off gradually as the saturation point is approached. The application is quite similar to that employed for silica gel, that is, the material is exposed to the air flow and after reaching about 75 per cent saturation is reactivated by removing the moisture adsorbed by means of applied heat. The actual scheme generally followed in the use of this material for continuous service varies somewhat from silica gel inasmuch as the material is placed in three units which are used consecutively for the different steps. These steps permit each unit to operate as follows:

a. In series with the preceding unit.
b. Alone.
c. In series with the following unit.

This plan allows for adsorption, reactivation and cooling the same as with silica gel.

Taking a single unit, when it is in the *a* step and operating with the preceding unit, the alumina absorbs approximately 25 per cent of the moisture in the air and takes up about 1.3 per cent of its weight of water. During the second step when it is operating alone, it takes up 100 per cent of the moisture in the air until the weight of the water absorbed is brought up to about 6.7 per cent; in the third step when the unit is operating with the following unit it extracts about 75 per cent of the moisture in the air until the water weight adsorbed comes up to about 10 per cent of the weight of the adsorber or a trifle over. The time allowable for reactivating is equal to the time occupied by the following (or second) unit adsorbing alone, plus the time when the second and third units are adsorbing in series, plus the time when the third unit is adsorbing alone, at the expiration of which time the original (or first) unit again will be required.

The temperature of air used for alumina reactivation is usually between 300 and 700 F and the air flow rate will have to be higher with the low temperature air than it will be with reactivating air of higher temperature. For example, air at 400 F for reactivating will, at 10 cu ft per hour, per

pound of alumina, require about 6 hours for reactivation. In the three unit system, after reactivation, the cooling of the activated alumina may be carried out with considerable rapidity by using dry air from the adsorption unit for circulation through the unit which has just completed reactivation and the final temperature of the unit before it goes back into service should be not over 200 F. As a basis for the amount of cooling air required each cubic foot of cooling air has been found capable of removing 2.2 Btu when heated from 85 to 200 F and still provide a sufficient margin of safety in operation.

REFRIGERATION

Air conditioning imposes requirements on refrigeration equipment not usually found in general cooling work, so that specially designed apparatus is often needed to replace that normally used for industrial cooling. Standard equipment can be adapted to meet air conditioning requirements but extreme care must be taken to determine the limits of its applicability.

In industrial or process cooling systems the load is fairly constant, noise in operation is not of paramount importance, space is available or relatively cheap, condenser water is not a source of worry, and the cooling system is to a great extent separate and independent of other mechanical equipment. By contrast air conditioning, especially as used for space cooling and comfort work in office buildings, theatres, and places of great density of population requires special consideration of all these factors. Space in public buildings is limited and condenser water is expensive. Noise interferes with the occupants and the cooling equipment must dovetail with the other air handling apparatus. Most important, the load fluctuates tremendously and is seasonal.

A complete discussion of the thermodynamic problems of refrigeration is given in the Refrigerating Data Book[2], 1932, so only a brief description of the cycle will be given here before the problems peculiar to air conditioning are considered.

The refrigeration system consists of three main parts, the evaporator, the condenser, and the compressor. Fig. 2 shows a diagram of the cycle. Heat is absorbed in the evaporator and released in the condenser. The compressor changes the level of the heat by taking it from a lower to higher plane. There are also many valves, accessories and special devices necessary for proper operation which vary somewhat with different types of cooling systems and different refrigerants.

Heat absorption is accomplished in the evaporator, or cooler, by maintaining a pressure sufficiently low to cause the refrigerant to boil at the temperature necessary to cool. The heat of ebullition is taken from the substance cooled, and the vaporized refrigerant is withdrawn by the compressor which raises the pressure to a point that permits the gas to liquify in the condenser when some readily available medium (usually water), is used to absorb the latent and super-heat. The liquid then returns to the evaporator through a pressure reducing valve and the cycle repeated. The temperature, and the corresponding pressure maintained in the evaporator is fixed by the temperature to which it is desired

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[2]Published by *American Society of Refrigerating Engineers.*

to cool the air. The temperature and pressure in the condenser are determined by the temperature and quantity of the condensing medium. The two pressures govern the size of the compressor and the amount of power required to drive it.

The heat released in the condenser is equal to the sum of the heat absorbed in the evaporator and the heat equivalent of the power required to drive the compressor, with small corrections for direct losses or gains from the surfaces exposed to the atmosphere.

When designing air conditioning systems, the capacity of equipment is fixed by selecting apparatus of sufficient size to maintain predetermined temperatures and humidities in treated spaces when arbitrarily established maximum atmospheric temperatures occur coincident with given population, lighting, power consumption, etc. These factors determine the maximum duty of the cooling system. The duty does not necessarily

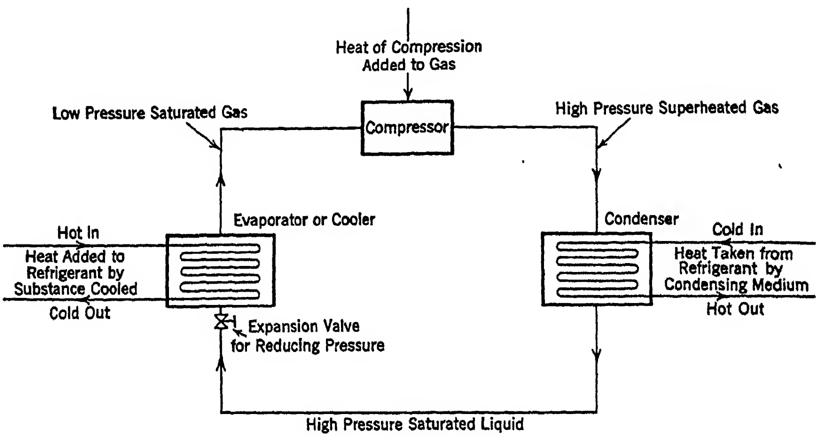


FIG. 2. TYPICAL REFRIGERATION DIAGRAM

determine the size or capacity of the refrigeration apparatus. The capacity is expressed in "tons ice melting effect" or in units equal to the absorption of the heat given up by one ton of ice at 32 F melting to water at 32 F in 24 hours. This is equivalent to heat absorption at a rate of approximately 200 Btu/minute.

After the maximum duty is determined, the other factors surrounding the installation must be investigated. The total heat to be removed by the cooling system has many sources, some substantially constant and others extremely variable. These sources can be roughly classified as follows, the first column indicating the order in size and second the order of variability:

1. Outside air supplied.	1. Fresh air supplied.
2. Population.	2. Transmission through walls.
3. Transmission through walls.	3. Light and power consumed.
4. Light and power consumed.	4. Population

By combining these two columns, a third grouping is obtained as follows:

1. Fresh air supplied.
2. Transmission through walls.
3. Population.
4. Light and power consumed.

In this last arrangement, the first two items are governed by atmospheric temperatures and therefore subject to tremendous fluctuations in size. As they generally form 40 per cent to 60 per cent of the entire maximum load, it is obvious that the duty of the cooling system will be much less than maximum most of the time.

A survey of Weather Bureau records indicates that maximum temperatures occur less than 5 per cent of the cooling period and also that the duration of peak conditions is never more than three or four hours.

Design of System

As previously mentioned, two factors control the size of the refrigeration system, the evaporator or suction, and the condenser or head, temperatures. With the knowledge that the system will operate most of the time with a load of not over 60 per cent of maximum, and that maximum demands will occur infrequently and only for short periods, some provision must be made to insure economical operation under average conditions. This can be done by overloading the machine under extreme demands and basing the design on normal or average loads. Flexibility in arrangement can be provided in several ways.

Variations in load change the efficiency of any machine and a refrigerating system can be costly and inefficient if improperly designed or operated. Fortunately, the trouble can be concentrated in the compressor and the problem relieved of many complications. It is comparatively easy to furnish condensers and evaporators to carry maximum load and so arranged that they will function properly at small demands. They affect the compressor performance to some extent but most of the compressor problems are in the machine itself.

Variations in load are usually effected by lowering the suction temperature and pumping a larger volume of gas per ton through a greater pressure range. This is possible because the latent heat of the refrigerant remains nearly constant throughout the small range used and the specific volume varies rapidly with change in pressure. As the compressor must remove the gas evaporated, the evaporator temperature fixes the displacement required. The objection to such a method is that the total power consumed remains nearly constant and the power per unit of cooling increases rapidly. Such operation is satisfactory as long as the load is kept within 10 per cent of the rating of the compressor but this condition does not commonly occur in air conditioning applications.

Operating Methods

It is possible to divide the entire refrigeration system into a number of small units, which will allow cutting in and out of compressors, condensers, etc., as the load fluctuates. This, however, is an expensive method as a number of small units are more expensive than one large unit. There is a certain amount of duplication of equipment necessary, which tends to increase the initial cost of the system and makes the fixed charges applicable to the operation of the air conditioning and cooling system, greater than necessary.

A second method of providing for economy of operation, is to have storage capacity which can be utilized during the peak period. A further reference to the Weather Bureau records, indicates that maximum conditions occur during the day for not more than three hours duration, and consequently, the refrigerating system can be run for a longer period at maximum efficiency, with tanks to store cold water or brine for supplementing the actual output of the refrigerating equipment when the load is more than the machine will carry. This situation brings complications. Storage tanks require space and extra apparatus, which increases the cost of the entire system, and further, it is difficult to determine what the size of the compressor should be, because of the other variables which enter the problem. Depending upon the availability of storage space, the compressor could be designed for any capacity above 50 per cent of the maximum load, so the smaller the compressor, the larger the storage space, and vice versa.

A third method, is to provide in the compressor itself, some means of reducing the capacity. This can be done by varying the speed (and consequently the displacement of the compressor), or by varying the displacement, either by a partial by-pass of the cylinder, or a clearance pocket in the head of the cylinder when reciprocating compressors are used. It might be assumed that the efficiency would remain practically constant. This is not correct, inasmuch as the machine friction remains constant with the by-pass or clearance pocket method, which raises the power required per ton of refrigeration developed. Also, the volumetric efficiency of the machine falls off rather rapidly when the clearance pocket or partial by-pass is used. By varying the speed of the compressor, the efficiency of the motor falls off as the speed is changed, the power output of the motor varies below maximum, and again, the compressor friction remains constant. Of the two methods, the clearance pocket, or partial by-pass of the cylinder is probably the more efficient, for general use.

Another method of operation is automatic starting and stopping of the refrigerating machine, with the automatic control designed to function as the load varies. This, however, is not considered good practice as mechanical troubles develop and the life of the system is impaired. While the equipment is kept in good condition, however, the machine will operate at maximum efficiency so long as it runs. The constant starting and stopping of large compressors is liable to cause the power factor to decrease if adequate allowance is not made.

All of the methods described are used from time to time.

The methods of varying the output of a refrigeration system which have been outlined, apply to the reciprocating type of compressor, although variations in the speed of the compressor to change the refrigerating output is common to all types of mechanical refrigeration.

There is a further method of controlling the compressor output which is particularly adaptable to the centrifugal type of machine. This is accomplished by varying the amount of condensing water used, with the fluctuation in demand load. Because of the characteristics of the centrifugal type of apparatus, as the condensing water quantity is reduced and the condensing temperature consequently raised, the discharge pressure of the centrifugal machine rises correspondingly and the horse power input to the machine falls off. While this reduces the total power input to the

machine, it does not necessarily reduce the power input per ton of refrigeration developed, as the power input does not drop with a rising discharge pressure as fast as the refrigerating effect produced. It is a method, however, which shows marked economies over the method generally used by the operating engineer, which is to lower the suction pressure in order to reduce the refrigerating output of the system.

Steam Jet System

So far the discussion has been confined to reciprocating, centrifugal, and rotary compressors. There is another type of compressor which, under certain circumstances, is desirable for use with air conditioning. Reference is made to the steam jet compressor. Fig. 3 shows a complete flow diagram of the system. The power used for compressing the refrigerant

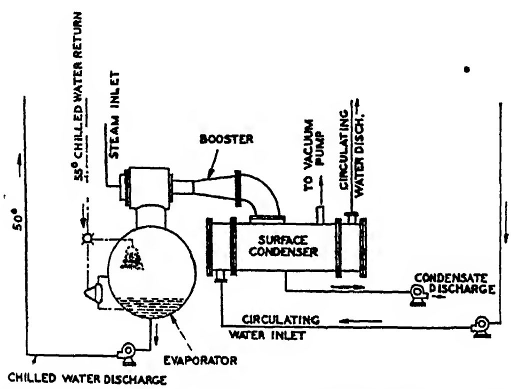


FIG. 3. DIAGRAM OF STEAM JET REFRIGERATION UNIT

is steam, taken directly from the boiler, thus eliminating the mechanical losses of manufacturing electric current. As the compression ratio between the evaporator and condenser under normal circumstances is large the mechanical efficiencies of the equipment are somewhat lower than those of the positive mechanical type of compressor; also the condensing water requirements are considerably greater, as both the refrigerant and the impelling steam must be condensed.

The steam jet system functions on the principle that water under high vacuum will vaporize at low temperatures, and steam ejectors of the type commonly used in power plants for various processes, will produce the necessary low absolute pressure to cause evaporation of the water.

Fig. 2 shows a typical water cooling application. The water to be cooled enters the evaporator and is cooled to a temperature corresponding to the vacuum maintained. Because of the high vacuum, a small amount of the water introduced in the evaporator is flashed into steam, and as this requires heat and the only source of heat is the rest of the water in the evaporator tank, this other water is almost instantly cooled to a temperature corresponding to the boiling point, determined by the vacuum maintained. The amount of water flashed into steam is an extremely small percentage of the total water circulated through the

evaporator, amounting to approximately 11 lb per hour per ton of refrigeration developed. The remainder of the water at the desired low temperature, is pumped out of the evaporator and used at the point where it is required.

The ejector compresses the vapor which has been flashed into the evaporator, plus any entrained air taken out of the water circulated, to a somewhat higher absolute pressure, and this mixes with the impelling steam on the discharge side of the jet. The total mixture of entrained air, evaporated water and impelling steam is discharged into a surface condenser at a pressure which permits the available condensing medium to condense it. The resulting condensate is removed from the condenser by a small pump, where it can be discharged to the sewer or returned to the system in the form of makeup water, or part of it may be returned to the boiler feed pump.

As the normal temperature of water required for air conditioning purposes is between 40 F and 50 F, with an average temperature of approximately 45 F, this type of water cooling is particularly desirable, as the efficiencies and operating costs compare very favorably with other types of refrigerating equipment, especially in view of the fact that the cooling apparatus is, as a general rule, less expensive to install.

Approximately three times as much condenser water is required for the steam jet cooling system as would be necessary with other types of mechanical refrigeration, but as the system can be designed with a large number of jets, each of which can be cut off as the load falls below maximum, constant refrigerating efficiency is maintained and frictional losses, volumetric inefficiencies, etc., are kept at a minimum.

The slight amount of air which may be entrained in the cooled water, is removed by a small secondary ejector which raises the pressure sufficiently so that the air can be discharged to the atmosphere. A small secondary condenser, of course, is necessary to condense the steam used in the secondary jet.

Steam jet refrigeration has an advantage, where cooling towers are used for supplying the condensing liquid, as there is a great saving in the amount of steam used per ton of refrigeration. As the outdoor weather conditions vary, and of course the load on the cooling system with it, the compression ratio between the condenser and evaporator can be reduced and less propelling steam used per ton of refrigeration developed. Roughly, in air conditioning work, mechanical compressors show a falling off of 30 to 40 per cent in the power input when using the most economical arrangement of compressors, as the load varies from 100 per cent to 25 per cent of the rated capacity; whereas with steam jet cooling equipment, the amount of steam required for producing the necessary refrigerating effect falls off in direct proportion to the load on the system or with cooling towers, the amount required falls off even more rapidly.

Compressors and Refrigerants

There are many different types of compressors, a number of refrigerants, different types of evaporators, condensers and arrangements of the cycle; each of which has its particular place and usage. There are four (4) different types of compressors, and in general, six (6) different refriger-

ants normally used for air conditioning purposes. The generally used compressors are of the following types:

1. Reciprocating compressors.
2. Centrifugal compressors.
3. Rotary compressors.
4. Steam Jet compressors.

The refrigerants in most general use are:

1. Ammonia.
2. Carbon Dioxide.
3. Dichlorodifluoromethane.
4. Dichloromethane.
5. Methyl-chloride.
6. Water Vapor.

The following is a brief discussion of the various types of compressors and their relationship to the refrigerants:

Reciprocating compressors can be used for any of the refrigerants listed, except water vapor and dichloromethane.

Centrifugal compressors can be used for dichloromethane or water vapor, and theoretically, for any of the other refrigerants, but the resulting loss in efficiency, with the higher pressure gases, limits the centrifugal compressor to the two refrigerants named.

The rotary compressors are generally used for methyl-chloride and dichlorodifluoromethane, because of their relatively low pressure and compression ratios.

The steam jet compressor is used only when water vapor is the refrigerant.

Reciprocating compressors are a thoroughly familiar piece of equipment and have been developed to a point where their efficiency is high and their operation very satisfactory. They generally operate at low speeds, and in large installations this fact makes them desirable for general use. They are of two types, the vertical and the horizontal type, either single or double acting. Reciprocating compressors are widely used in ordinary refrigeration work and they can be used with more refrigerants than other types of compression units. For instance, when carbon dioxide is used as the refrigerant, the reciprocating compressor is used because of the extremely high pressures and relatively high ratio of compression.

Centrifugal compressors are usually built in two or more stages where the compression ratio is high and their design follows closely that of any other centrifugal equipment, such as general service pumps and fans.

Rotary compressors may be used for medium or low pressure refrigerants, with small compression ratios.

Steam jet compressors which have recently entered the field, cannot be used economically for water temperatures much below 40 F. They are simple and compact, have no moving parts and produce practically no vibration. Further, water is a cheap and inexhaustible refrigerant and does not need to be used in an enclosed system in the same way as other gases.

The source of condensing water to some extent governs the type of refrigerant used. If condensing water is available at temperatures of not more than 70 to 75 F any of the refrigerants mentioned can be used economically, but if the available condensing water temperature is above 80 deg, carbon dioxide becomes uneconomical as its critical temperature is approximately 88 F. A condensing water temperature over 80 deg makes the power required for compression high. All refrigerants have critical temperatures and pressures sufficiently high so that their efficiency is not materially affected by the condensing water temperatures, except in so far as this temperature affects the compression ratio.

Steam jet cooling systems can use water up to 85 F, or even slightly higher.

The applicability of the various refrigerants is interesting. Carbon dioxide is limited by the condensing water temperature, the power consumption is slightly higher than other refrigerants, and the pressures are three to four times that of ammonia.

Ammonia, probably the best known refrigerant, has the disadvantage of being toxic, and under certain circumstances, explosive, corrosive, and irritating, even in small quantities in the atmosphere.

Dichloromethane operates at pressures below that of the atmosphere, and it is to some extent toxic.

Dichlorodifluoromethane under normal circumstances is non-toxic, non-irritating, and non-explosive, but under high temperatures breaks down into several obnoxious, poisnous components.

Methyl-chloride, under certain conditions, is explosive and slightly toxic.

The steam ejector water vapor system has none of the disadvantages of toxicity, explosiveness and corrosiveness encountered in the other refrigerants, but the system operates at less than atmospheric pressure. This, however, is not an important factor as there are no moving parts in the compressor and the possibility of inleakage of air is remote as all of the equipment can be welded air and watertight. The supply of water is inexhaustible, and as a refrigerant, the makeup cost is negligible. The same boiler equipment for heating in winter and for cooling in summer can be used.

Motors

The motors used for driving compressors can be roughly classified in three groups; synchronous, multi-speed, or variable speed. Further information on motors may be found in Chapter 17.

Coolers

The types of coolers used in connection with air conditioning work fall into three general groups. The *first* is the direct cooling of water; the *second*, direct cooling of air; and the *third*, cooling of brine for circulation in a closed system, which can cool either water or air. In the order named, the cooling of water sprayed in the dehumidifier is the type most generally in use and can be accomplished in several ways. One method is to install direct expansion coils in the spray chamber so that the water sprayed in the air comes in direct contact with the cooling coils. Another common and rather efficient method of cooling spray water, is to use a baudelot type of heat absorber where the water flows over direct expansion coils at a rate sufficiently high to give efficient heat transfer from water to refrigerant, and used in connection with a storage tank or collecting pan insures good operation of the refrigeration equipment if proper attention is given to the load on the system.

Another type of spray water cooler is the shell and tube heat exchangers; the refrigerant being expanded into a shell enclosing the tubes through which the water flows. The velocity of the water in the tubes affects the rate of heat transfer, and as the refrigerant is in the shell, completely surrounding the tubes at all times, good contact and a high rate of heat transfer is insured. The disadvantage of such a system is that with the falling off of load on the compressor, the suction temperature or the temperature in the evaporator drops and there is a possibility of freezing the water in the tubes, which, of course, would split the tubes and allow the refrigerant to escape into the water passage. This danger can be eliminated by complete automatic safety devices.

Another system of cooling spray water is to submerge coils in the spray collecting tank, or in a separate tank used for storage. The heat transmission through the walls of the coils, however, is low and a great deal more surface is required than any other type of cooler. However, with large storage tanks, this type of cooling can be utilized to advantage.

In the second group of coolers mentioned, that of direct cooling of air, there is only one type to be considered; the coil type with the refrigerant in the coil and the air passing over it. Cooling depends upon convection and conduction for removing the heat from the air. The type of coil used for cooling can be either smooth or finned; the finned coil being more economical in space requirement than the smooth coils.

The third group of coolers, which are sometimes called indirect coolers, where brine is cooled by the refrigerant and the resulting cold brine used to cool either air or water, introduces several other considerations. It is not the most economical from a power consumption standpoint, as it is necessary to cool the brine to a temperature sufficiently low so that there is an appreciable difference between the average brine temperature and the substance being cooled. This requires that the temperature of the refrigerant must be still lower, and consequently the amount of power required to produce a given amount of refrigerant increases due to the higher compression ratio, but there are other considerations which make such a system desirable. In the first place, where a toxic refrigerant is undesirable or cannot be used, due to fire or other risks, especially in densely populated areas, the brine can be cooled in an isolated room or building, and the brine then circulated through the air conditioning equipment in perfect safety; the brine being used to cool the water or air, without any possibility of direct contact between the air and refrigerant.

When an indirect system of cooling is used, it will be found that the heat transfer rate of the water cooler is considerably higher as a general rule than a direct exapansion cooler for the same requirements. With direct expansion interchangers, it is almost inpossible to keep the entire system flooded with liquid, whereas with brine interchangers, the cooling medium completely fills the space of the interchanger and perfect contact is insured.

There is one other type of water cooling which might be mentioned, and that is the use of ice. Its application is limited because of the cost of ice, although the efficiency of cooling is higher than any other water cooling system. The word "water cooling" is used advisedly in that the direct cooling of air by ice is, while not impossible, rather impractical. It might be said that ice coolers are economical for systems requiring a maximum rate of 20 tons ice melting effect per 24 hours where the load fluctuates considerably, and it is possible to introduce ice only as it is required to cool water. The most general method of cooling water with ice is to spray the water over the surface of the ice, insuring as much contact as possible and approximating the same performance as the baudelot type of cooler. Because of the large fluctuations in load in the air conditioning system, the higher cost of refrigerating effect when ice is used, is offset by the fact that there are no motor and condenser inefficiencies under partial load, and the cost of the mechanical refrigeration equipment for the small system being so much higher per unit of effect, the fixed charges are small enough to overbalance the extra cost of the ice.

Condensers

Condensers are usually either double pipe or shell and tube type. Shell and tube condensers are almost identical with coolers. Double pipe condensers are arranged so that water passes through the inner of two con-

centric pipes and refrigeration through the annular space in the outer pipe. Where possible, the flow of refrigerant and condensing water should be counter flow to maintain maximum temperature differences.

The amount and temperature of the condensing water determines the condensing temperature and pressure, and indirectly the power required for compression. It is, therefore, necessary to strike a balance so that the quantity of water insures economical compressor operation. As part of the condenser, or attached to it, there must be storage space for liquid refrigerant.

The installation of all equipment should be made accessible for inspection, repair, and cleaning. Both the coolers and condensers should have space for pulling tubes.

In connection with air conditioning equipment and the refrigeration system used there is a decided tendency to conserve the water in the city mains and most large cities are restricting the use of this water. In order to use air conditioning systems and refrigeration equipment, it is often necessary to install cooling towers. The cooling towers, unfortunately, produce the highest temperature condensing water at the time when the load on the system is greatest, so that the refrigeration equipment must be designed to meet not only the maximum load at normal conditions, but the maximum load at abnormal condensing water temperatures. If properly designed with the flexibilities mentioned before, this makes little difference in the efficiency of operation throughout the year, except at those times when the condensing water temperature is highest. As this only occurs for 5 per cent of the entire cooling period it can be disregarded as a factor in establishing yearly operating costs.

The cooling tower has a certain advantage over the use of water from the city mains, in that the temperature of the condensing water varies directly with the outdoor temperature and as pointed out, the refrigeration load also varies with this temperature. Certain economies are possible when a cooling tower is used, which cannot be achieved by the use of condensing water from city mains, even where the city water temperature is extremely low. Normally, the lowest city water temperature met during the summer months is from 65 to 70 F. This temperature range takes place for the entire cooling period, regardless of what the outdoor temperatures are. With the cooling tower, the temperature of the condensing water may rise to 80 to 85 F under maximum conditions, but under less than maximum conditions, the temperature of the water off the cooling tower drops considerably, and it has been established that 50 per cent of the time the outdoor wet-bulb temperature varies from 60 to 70 F and the cooling tower water therefore, for the same periods, varies from 65 to 75 F. When the outdoor wet-bulb temperature drops below 60, which occurs approximately 30 per cent of the time, the condensing water temperature is still lower. The cost of water used for condensing is negligible as the only water required is that used for makeup due to the loss by evaporation in the cooling tower itself. See also Chapter 11.

Chapter 11

HUMIDIFYING AND DEHUMIDIFYING EQUIPMENT

Air Washers, Selection, Scrubber and Eliminator Surface, Distributing Plates, Materials of Construction, Performance, Heating Spray Water, One and Two Bank Humidifiers, Power Requirements, Nozzles and Wash Water Used, Humidifier Efficiency, Flooding Surface Type Humidifiers, Dehumidifiers, Cooling Medium, Atmospheric Water Cooling Equipment, Theory of Atmospheric Cooling, Atmospheric Cooling Reverse of Humidifying, Factors Affecting Atmospheric Water Cooling, Co-ordinating of Equipment, Wet-Bulb Temperature of Design, Quantity of Cooling Water, Cooling Ponds, Spray Ponds, Spray Retention, Growths in Spray Ponds, Spray Cooling Towers, Natural Draft Deck Cooling Towers, Wind Velocities for Towers, Mechanical Draft for Towers, Indoor Cooling Towers, Make-Up Water, Winter Freezing

AN air washer is essentially a chamber in which air is brought in intimate contact with water, the object being (*a*) to wash the air or (*b*) to regulate the moisture content of the air and at the same time wash it. The air comes in contact with the water by passing it through water sprays or by passing it over surfaces wetted by a continuous flow of water; hence the classification: spray, scrubber, and combination spray and scrubber type washers.

A washer chamber may be constructed of wood, or stone, but it is most often constructed of sheet metal. The lower portion of it is specially designed as a tank to receive the water dropping through the chamber and to serve as a reservoir from which the water may be recirculated.

It is desirable that air leaving a washer contains no water in suspension. For this reason eliminators are provided at the washer outlet. These may be in the form of plates or baffles upon which the free moisture is deposited as the air is deflected through several changes from its original direction of flow. In some washer units steel wool filter sections serve as eliminators. However, specially designed plates are used more generally than other devices because they offer the least resistance to the flow of air, while still performing effectively the function of free moisture elimination. They also have the advantage of acting as scrubber surfaces when flooded.

It is essential to uniform performance in a washer, that air enter evenly distributed over the washer inlet. To insure this, a perforated plate or eliminator plates are installed at the inlet. Eliminator plates are now more generally used. They serve a second purpose in preventing the escape of spray through the washer inlet.

Water is supplied to scrubber type units through flooding nozzles. The capacity of these nozzles varies with the manufacturer although a fair

value of 5 gpm may be used. The nozzles are spaced on one foot centers across the top of the washer over the scrubber plates.

Water is supplied to spray type units through atomizing nozzles generally arranged in banks across the washer. The nozzles spray either in the direction of the air flow, that is, downstream, or against the air flow, or upstream. Nozzle capacities varies with the manufacturer, from 1-½ to 2 gpm at a water pressure of about 25 lbs per square inch which pressure is required for effective atomization. The spacing of spray nozzles is determined by the water requirements of the particular installation. A spray type washer may contain one, two or three banks of nozzles depending upon its application.

When an air washer is used for cleaning air it removes impurities and dusts. In general it does not function as efficiently in this service as a filter. For non-microscopic soluble dust its efficiency averages about 50 per cent, unless the concentration of dust is high. Its effectiveness in removing greasy microscopic dust is practically negligible as is also its deodorizing ability.

When a washer is used to regulate the moisture content of air it adds moisture to (humidifiers) or removes moisture from (dehumidifiers) the air to achieve the desired moisture content. (See also Chapter 3).

When air passes through a washer wherein water is circulated without the addition or removal of heat, the air tends to become saturated at its entering wet-bulb temperature. What occurs here is partial or complete adiabatic saturation. The total heat content of the air is unchanged, inasmuch as the dry-bulb temperature of the air drops in proportion to the amount of additional water evaporated. This action is also known as evaporative cooling. A measure of the washer's effectiveness under these conditions is its saturating efficiency which is equal to the drop in dry-bulb temperature in per cent of the entering wet-bulb depression. Other things being equal, the saturating efficiency of a spray type washer is a function of the number of spray banks and the direction in which they spray. The following table gives a general comparison:

3 banks—2 upstream—1 downstream	100% saturation efficiency
2 banks—2 upstream	95% saturation efficiency
2 banks—1 upstream—1 downstream	85% saturation efficiency
1 bank —upstream	80% saturation efficiency
1 bank —downstream	65% saturation efficiency

When air passes through a washer wherein the circulated water is either cooled or heated before being returned to the spray chamber, a heat interchange between the air and water occurs, and the air tends to become saturated at the temperature of the leaving water. The extent to which the leaving air and leaving water temperatures approach each other is an index to the effectiveness of the washer under the operating conditions. The total heat absorbed by the water in the process equals the total heat given up by the air or the heat given up by the water equals the heat absorbed by the air. Depending on whether the moisture content of the air is increased or decreased during the operation, humidification or dehumidification occurs. Heat will be added to or removed from the air as the water supplied is of a higher or a lower temperature than the wet-bulb temperature of the entering air.

For dehumidifiers the ratio of the difference between the leaving wet-bulb and the leaving water to the difference between the entering wet-bulb and the entering water may be figured as follows:

3 banks—1 downstream—2 upstream	0%
2 banks—2 upstream	5%
2 banks—1 upstream—1 downstream	15%
1 bank —upstream	20%
1 bank —downstream	35%

Humidifiers may be figured on the same basis as dehumidifiers; the leaving water temperature, of course, will be higher than the wet-bulb temperature of the leaving air.

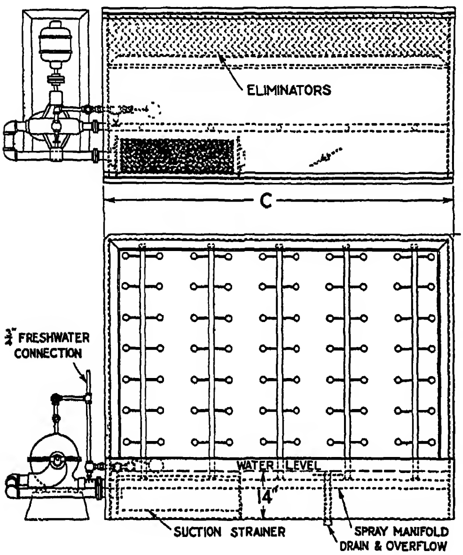


FIG. 1. TYPICAL SINGLE BANK AIR WASHER

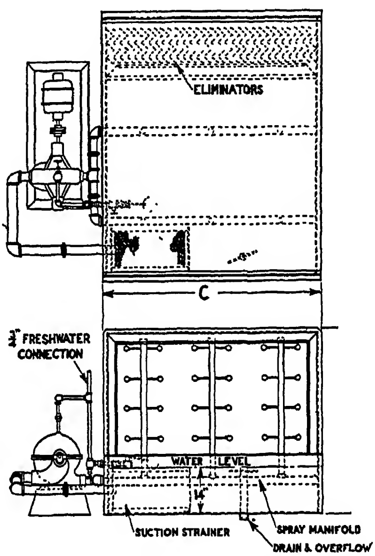


FIG. 2. TYPICAL TWO BANK AIR WASHI

The problem of cooling or heating the circulated water before returning it to the washer chamber is external to the unit. It will suffice here to note that heating is generally accomplished by passing the water through closed hot water heaters or by injecting steam into the water circuit; cooling, by passing the water through closed coolers or over refrigerating coils in a baudelot chamber. Often in a cooling and dehumidifiying application, the refrigerating coils are located within the washer chamber.

Washers are sometimes arranged in two or more stages to cool through long ranges or to increase the overall efficiency of heat transfer between air and the cooling or heating medium (water, brine, etc.) A multi-stage washer is equivalent to a number of washers in series arrangement. Each stage is in effect a separate washer.

Usually the catalog capacity of a washer is expressed in cubic feet of air per minute and is based upon an air velocity of 500 feet per minute through the gross cross sectional area of the unit above the water level in its tank. At this rating spray type washers handle about 2-½ gpm of

water per bank per square foot of area, that is, about 5 gpm per bank per 1000 cfm. These proportions of air, water, area, and velocity may be departed from to meet the needs of some particular job, but certain limiting relationships should be observed. Two of the more important items are:

a. Choose a washer for air velocities above approximately 300 fpm and below approximately 600 fpm. Velocities outside this range are likely to result in faulty elimination of entrained moisture.

b. When a high saturating efficiency is required, select a two or three bank spray type unit, having a total water capacity of not less than 15 gpm per 100 cfm.

The area of a washer may be dictated by space limitations outside the washer, such as headroom, or by space requirements inside washer, such as face area needed by a bank of cooling coils. The length of a washer is determined by the number of spray banks, or scrubber plates, and if cooling coils are installed in the unit, by the number of banks of coils.

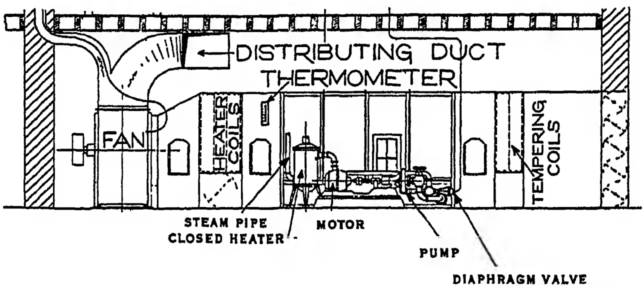


Fig. 3. Air Washer with Spray Water Heating Arrangement

Roughly, a spray space of about 2 ft-6 in. in length is required for each bank of sprays, (the *leaving* eliminators require about 1 ft-6 in., *entering* eliminators about 1 ft.)

The resistance to air flow through an air washer varies with the type eliminators, number of banks of sprays, direction of spray, type of scrubber plates, and, if cooling coils are located in unit, by their size and type. Washers should be selected to limit static resistances below 0.50 in.

Power Requirements

The approximate power requirement for passing 10,000 cfm of air through a humidifier of the spray type by a fan of 78 per cent mechanical efficiency is given in Table 1, this being the fan brake horsepower for various velocities and static pressure losses. Allowance should be made for variations in static pressure due to the use of different diffuser plates or inlet louvres and for variations in fan efficiencies.

ATMOSPHERIC WATER COOLING EQUIPMENT

To successfully operate a refrigerating plant or a condensing turbine, the heat from the compressed refrigerant or the discharged steam must be removed and dissipated. This is accomplished ordinarily by first transferring the heat of the gas to water in a heat exchanger. If the plant is

situated on the banks of a river or lake, an intake may be had upstream or at a considerable distance from the discharge, to prevent mixing of the heated discharged water with the inlet water. If the source of water is a city supply or well water, the discharge water may be run into the nearest sewer or open water way. Lacking an unlimited water supply, or in cases where city water is too expensive or where the water available contains dissolved salts which would quickly form scales on the heat-exchanging apparatus, it is necessary to recirculate the water, and to cool it after each passage through the heat-exchanger by exposure to air in an atmospheric water cooling apparatus.

As air has a capacity for absorbing heat from water when the wet-bulb temperature of the air is lower than the temperature of the water with which it is in contact, the rapidity with which this transfer of heat occurs depends upon (1) the area of water in contact with the air, (2) the relative velocity of the air and water, and (3) the difference between the wet-bulb temperature of the air and the temperature of the water. Because the changes in rate do not occur in direct proportion to changes in the governing factors, data on the performance of atmospheric water cooling equipment are largely empirical.

TABLE 1. APPROXIMATE FAN BRAKE HORSEPOWER

Requirements for passing 10,000 cfm of air through humidifiers at various velocities and static pressures. Mechanical efficiency of fan—78 per cent.

Velocity fpm	30 Deg Eliminators Spaced on 1-½ in. Centers		45 Deg Eliminators Spaced on 2-¼ In. Centers	
	Static Pressure In. Water	BHP	Static Pressure In. Water	BHP
500	0.20	0.40	0.40	0.80
550	0.24	0.48	0.48	0.97
600	0.29	0.58	0.58	1.15
650	0.34	0.68	0.68	1.35

As the heat content of the air increases, its wet-bulb temperature rises. (See Chapter 1). Because it is impractical to leave the air in contact with water for a long enough time to permit the wet-bulb temperature of the air and the temperature of the water to reach equilibrium, atmospheric water cooling equipment aims to circulate only enough air to cool the water to the desired temperature with the least possible expenditure of power.

Cooling Towers

In an air washer, humidifier or dehumidifier, the air is first conditioned by water to change its moisture and temperature, and it is then sent to the place where it is to be used. In water cooling equipment the temperature of the water is reduced by air, and the cooled water is carried to its point of usage. In the air washer, an excess of water is used to condition a fixed quantity of air, while in water cooling equipment, an excess quantity of air is used to cool a fixed quantity of water.

Both types of equipment have a common basis of design, however, in that the size of the equipment is determined by the quantity of air that

must be handled. With the air washer, the size of the equipment is fixed by the quantity of air to be conditioned, and the amount of conditioning is controlled by the quantity and temperature of the water supplied and its method of application. With water cooling apparatus, its size and the quantity of air required bear no direct relation to the quantity of water being cooled, but vary through a wide range for different services and conditions.

Sizes of Equipment

Assuming a definite quantity of water to be cooled, the size and design of atmospheric cooling equipment is affected by the following factors:

1. Temperature range through which the water must be cooled.
2. Number of degrees above the wet-bulb temperature of the entering air to which the water temperature must be reduced.
3. Temperature of the atmospheric wet-bulb at which the required cooling must be performed.
4. Time of contact of the air with the water. (This involves height or length of the apparatus and velocity of air).
5. Surface of water exposed to each unit quantity of air.
6. Relative velocity of air and water.

TABLE 2. CONDENSER DESIGN DATA

Gas	Maximum Pressure Desired in Condenser	Gas Temperature in Condenser F	Leaving Hot Water Temperature F	
			Best Design	Average Design
Steam	28 in. vacuum	99.7	97	93
Steam	27 in. vacuum	114.3	110	105
Steam	26 in. vacuum	126.0	120	114
Ammonia	185 lb gage head pressure	96.0	92	88
Carbon dioxide	1030 lb gage head pressure	86.0	83	81
Methyl chloride	102 lb gage head pressure	100.0	96	92
Dichlorodifluoromethane	117 lb gage head pressure	100.0	96	93

Items 1, 2, and 3 are established by the type of service and geographical location, while items 4, 5, and 6 depend upon the design of the equipment.

The establishment of a proper cooling range depends upon:

1. Type of service, (refrigerating, internal combustion engine and steam condensing).
2. Wet-bulb temperature at which the equipment must operate satisfactorily.
3. Type of condenser or heat-exchanger used.

Because the design of an entire plant is usually affected by the quantity and temperature of the cooling water supply, plants should be designed for cooling water conditions which can be most efficiently attained. The first consideration is usually the limiting temperature of the plant. For example, if an ammonia compressor refrigerating plant is to be designed for 185 lb head pressure as a normal maximum, the limiting temperature of the ammonia in the condenser is 96 F. Should the ammonia temperature go above this figure the head pressure will exceed 185 lb and power con-

sumption increases. To obtain this head pressure, the temperature of the circulating water leaving the condenser must always be less than 96 F by an amount depending upon the size and design of the condenser, the quantity of water being circulated, and the refrigerating tonnage being produced. A condenser having a large surface per ton of refrigeration may be designed to operate satisfactorily with the leaving hot water temperature within 3 deg or 4 deg of the ammonia temperature corresponding to the head pressure, while a small condenser might require a 10 deg difference.

Table 2 lists several gases with data as to the temperature and pressures for which commercial condensers are designed. Internal combustion engines have limiting hot water temperatures of 125 F to 140 F. The cooling of such fluids as milk or wort has variable requirements and is usually done in counter-flow heat-exchangers in which the leaving circulating water is at a much higher temperature than is the leaving fluid.

The temperature range, once the hot water temperature is approximately known, depends upon:

1. Maximum wet-bulb temperature at which the full quantity of heat must be dissipated.
2. Efficiency of the atmospheric cooling equipment considered.

Design Wet-Bulb Temperatures

The maximum wet-bulb temperature at which the full quantity of water must be cooled through the entire range is never, in commercial design, the *maximum* wet-bulb temperature ever known to exist at the location nor the *average* wet-bulb temperature over any period. The former basis would require atmospheric cooling equipment several times greater than normal size, and the latter would result during a large part of the time, in higher condenser water temperatures than those for which the plant was designed. For instance, the maximum wet-bulb temperature recorded in New York City is 88 F, and the July noon average for 64 years is close to 68 F. Yet in the years 1925 to 1931, inclusive, there were but 6 hrs per year, when the wet-bulb temperature reached 80 F or more, and there were 975 hours in the average summer (June to September, inclusive) when the wet-bulb temperature was 68 F or above. As these 975 hours represent a third of the summer period, cooling equipment based upon the noon average July wet-bulb of 68 F would be inadequate. Commercial practice is to choose a wet-bulb temperature for refrigeration design purposes which is not exceeded during more than 5 to 8 per cent of the summer hours (75 F for New York City), with somewhat lower requirements for steam turbines and internal combustion engines. This difference is made because the heaviest load on a refrigerating plant is coincident with high wet-bulb temperatures, whereas the heaviest electric power demand occurs either in the winter or after nightfall in summer, when the wet-bulb temperature is low. Table 1, Chapter 8, shows safe design wet-bulb temperature which will not be exceeded more than 8 per cent of the time in an average summer.

Knowing the hot water temperature and the wet-bulb temperature for which the equipment must be designed, the cold water temperature must be chosen to place the requirement within the efficiency range of the type of atmospheric water cooling apparatus to be used. Efficiency of atmos-

pheric water cooling apparatus is expressed as the percentage ratio of the actual cooling range to the possible cooling range. Since the wet-bulb temperature of the entering air is the lowest temperature to which the water could possibly be cooled this is:

Percentage cooling efficiency of atmospheric water cooling equipment =

$$\frac{(\text{hot water temperature} - \text{cold water temperature}) \times 100}{\text{hot water temperature} - \text{wet-bulb temperature of entering air}}$$

Efficiencies of various types of atmospheric water cooling apparatus vary through wide limits, depending upon air velocity, concentration of water per square foot of area, and the type of equipment. The commercial range of efficiencies is given in Table 3 although unusual designs may operate outside these ranges.

TABLE 3. EFFICIENCY OF ATMOSPHERIC WATER COOLING EQUIPMENT

EQUIPMENT	COOLING EFFICIENCY—PER CENT		
	Minimum	Usual	Maximum
Spray Ponds	30	45 to 55	60
Spray Towers	40	45 to 55	60
Natural Draft Deck or Atmospheric Towers	35	50 to 70	90
Mechanical Draft	35	55 to 75	90

From consideration of the factors which include the cooling range and design wet-bulb temperature, the quantity of water required can be calculated from the amount of heat to be dissipated. The normal amounts of heat to be removed from various parts of the cooling equipment are:

Compressor refrigeration	220 to 270 Btu per minute per ton
Condenser turbine	950 to 980 Btu per pound of steam
Steam jet refrigerating appartus	1030 to 1150 Btu per pound of steam
Diesel engine	2800 to 4500 Btu per horsepower

Cooling Ponds

A natural pond is often used as a source of condensing water. The hot water should be discharged close to the surface at the shore line, as natural air movement over the surface of the water will cause evaporation and carry away heat. Because increased density due to the loss of heat causes the cooled water to sink to the bottom of the pond, the suction connection for intake water should be placed as far below the surface as possible, and at as great a distance from the discharge as practicable.

Spray Cooling Ponds

The spray pond consists of a basin, above which nozzles are located to spray water up into the air. Properly designed spray nozzles break up the water into small drops, but not into a mist because the individual drops must be heavy enough to fall back into the basin and not drift off. The water surface exposed to the air for cooling is the combined area of all the small drops. Since the rate of heat removal by atmospheric water cooling is a function of the area of water exposed to the air, the difference in

temperature between the water and the wet-bulb temperature of the air, the relative velocity of air and water, and the duration of contact of the air with the water, a much larger quantity of heat may be dissipated in a given area with the spray pond than with the cooling pond, because of (1) the speed with which the drops travel as they are propelled into the air and fall back into the water basin, (2) the increased wind velocity at a point above the surrounding structures or terrain, (3) the increased volume of air used, and (4) the vastly increased area of contact between air and water.

Spray pond efficiencies are increased by (1) elevating the nozzles to a higher point above the surface of the water in the basin, (2) increasing the spacing between nozzles of any one capacity, (3) using smaller capacity nozzles, to decrease the concentration of water per unit area, and (4) using smaller nozzles and increasing the pressure to maintain the same concentration of water per unit area. Usual practice is to locate the nozzles from 3 ft to 6 ft above the edge of the basin, to supply from 5 lb to 12 lb pressure at the nozzles, using nozzles spraying from 20 gpm to 60 gpm each and spacing them so the average water delivered to the surface of the pond is from 0.1 gpm per square foot per minute in a small pond to 0.8 gpm per square foot per minute in a large pond.

Increasing the pressure, spacing the nozzles farther apart, or increasing the elevation of the nozzles will increase the cross section of spray cloud exposed to the air, and therefore increase the quantity of air coming in contact with the water. Best results are obtained by placing the nozzles in a long relatively narrow area located broadside to the wind.

Spray ponds may be located on the ground if they have an earthen or a concrete basin, or they may be placed on roofs having special waterproof roofing. To prevent excessive drift loss, or the carrying of entrained water beyond the edge of the pond by the air on the leeward side, louver fences are required for roof locations and for those ground locations where space is so restricted that the outer nozzles cannot be located at least 20 ft to 25 ft from the edge of the basin. Such fences usually are constructed of horizontal louvers overlapping so the air is forced to turn a corner in passing through the fence, and the heavier drops of water are thrown back, owing to their inertia. The louvers also restrict the flow of air, particularly at the higher wind velocities, and thus further reduce the possibility of water being carried off. The height of an effective fence should be equal to the height of the spray cloud. Louver boards are preferably of red gulf cypress or California redwood supported on cast-iron, steel or wood posts. Where building ordinances forbid the use of combustible materials, sheet metal is customarily used.

Algae formations may be a considerable nuisance in a spray pond. Such growths are killed by the periodic addition of potassium permanganate to the pond water. Addition of the dissolved chemical should be made until the water holds a faint pink color for at least 15 min.

Spray Cooling Towers

Where not more than 30,000 Btu per minute are to be dissipated, the spray cooling tower is a satisfactory apparatus. The word *tower* in this connection is somewhat of a misnomer as the apparatus is essentially a

narrow spray pond with a high louver fence. As usually built, the nozzles spray down from the top of the structure and the distance from the center of the nozzle system to the fence on either side is not more than half the distance that the nozzles are elevated above the water basin. Heights range from 6 ft to 15 ft and the total width of a structure is not usually greater than its height. Spray cooling towers occupy less space on small jobs than spray ponds of equivalent capacities because the towers have a capacity of from 0.6 gpm to 1.5 gpm per square foot of tower area. The louvers are continually wet, and so add to the surface of water exposed to the cooling air.

Natural Draft Deck Type Towers

In past years most of the atmospheric water cooling on refrigeration work has been done with natural draft deck type towers, which are also referred to as *wind* or *atmospheric* towers. These towers consist of heavy wooden or steel framework from 15 ft to 80 ft high and from 6 ft to 30 ft wide, having open horizontal lattice-work platforms or decks at regular intervals from top to bottom, and a catch basin at the foot. The hot water is distributed over the upper part of the structure by means of troughs, splash heads, or nozzles, and it drips from deck to deck down to the basin. The object of the decks is to arrest the fall of the water so as to present efficient cooling surfaces to the air, which passes through the tower parallel to the decks. The decks also add to the area of water surface exposed to the air, but since they furnish a resistance to air flow, too many decks are a detriment.

To prevent the loss of water on the leeward side of the tower, wide splash boards are attached at regular intervals from top to bottom. These boards or louvers extend outward and upward, and in most designs the top edge of each louver extends above the bottom edge of the one above it.

Efficiency of a deck tower is improved, within limits, by increased height, increased length, or increased width. The first two increases the area of water exposed to the wind, and the latter increases the time of contact of the air with the water.

Wind Velocities on Natural Draft Equipment

Since natural air movement is the prime requirement for a deck type tower, spray cooling tower, or spray pond, the apparatus must be designed to produce the desired cooling on days when the wind velocity is below average when the wet-bulb temperature is at the maximum chosen for design, and when the plant is operating at full load. The apparatus must also, for best results, be located with its longest axis at right angles to the direction of the prevailing hot weather breeze. Table 1 Chapter 8, gives the average summer wind velocities and directions in representative cities. Natural draft cooling equipment should be designed to operate properly with *not more than one-half* of the *average* wind velocity, and in no case should it need a wind velocity of more than 5 mph. It is obvious that natural draft towers and other natural draft equipment must be so located that they are not obstructed by trees, buildings, or other wind obstructions.

Mechanical Draft Towers

Mechanical draft towers usually consist of vertical shells, constructed of wood, metal or masonry, in which water is distributed uniformly at the top and falls to a collecting basin at the bottom. The inside of the tower may be filled with wood checker-work over which the water drips, or the water surface may be presented to the air by filling the entire inside of the structure with spray from nozzles. Air is circulated through the tower from bottom to top by forced or induced draft fans. Since the air flows counter to the water, the air is in contact with the hottest of the water just before leaving the top of the tower, and each unit of air picks up more heat than a similar unit would on natural draft equipment, so the mechanical draft tower cools water by using less air than the other types of equipment need. As movement of the air through the towers is obtained by power-consuming fans, it is essential that the air used be reduced to a minimum so as to secure the lowest possible operating cost.

The efficiency of a mechanical draft tower is increased by increasing height, area, or air quantity. Increasing the height increases the length of time the air is in contact with the water without affecting seriously the fan power required, but it increases the pumping power needed. Increasing the area while maintaining constant fan power increases the air quantity somewhat and because of louvered velocities it increases the time this air is in contact with the water. The surface area of water in contact with the air is increased in both cases. Increasing the air quantity decreases the time the air is in contact with the water, but, since a greater quantity is passing through, the average differential between the water temperature and the wet-bulb temperature of the air is increased, and this speeds up the heat transfer rate. Increased air quantities are obtained only at the expense of increased fan power, which increases approximately as the cube of the air quantity. Air velocities through mechanical draft towers vary from 250 fpm to 600 fpm over the gross area of the structure.

Inside Mechanical Draft Water Cooling Equipment

Mechanical draft water cooling equipment may be set up inside buildings, where it usually draws its air supply from the general space in which it is installed, and discharges its exhaust air through a duct to the outside. Indoor cooling towers may be either of the wood-filled or the spray-filled type. In many cases where little height but considerable area is available, water is cooled in a spray-filled structure similar to an air washer, with the air passing horizontally through the apparatus and being discharged through a duct to the outside. Such apparatus does not have the counter flow advantage of the vertical mechanical draft water cooling equipment, and therefore requires a much larger excess of air for proper operation. Air velocities and operating powers are considerably above those required by vertical mechanical draft water cooling equipment.

Make-up Water

Since the atmospheric water cooling equipment performs its functions chiefly by evaporating a portion of the water in order to cool the remainder, there is a continual drain on the quantity of water in the system,

and this loss must be replaced. Approximately 1 gal of water is lost for every 1000 gal of water cooled per degree of cooling range; so if 1000 gpm of water are cooled through a 10 deg range, 10 gpm of water will be required to replace evaporated water. Replacement supply is usually regulated by a float control valve. Because the evaporation of the water leaves behind the salts which the water contained, high concentration of salts may make chemical treatment of the make-up water necessary to avoid excessive deposits in the condensers.

Winter Freezing

If atmospheric water cooling equipment is operated in freezing weather, the water may be cooled below freezing temperature so ice forms and collects until its weight causes damage. To obviate freezing during continued operation, the efficiency of the apparatus may be lowered. This is done on the spray pond and the spray cooling tower by reducing the quantity of water fed to the apparatus, thereby lowering the pressure at the nozzles and increasing the size of the drops produced. On the deck

TABLE 4. COMPARISON OF VARIOUS TYPES OF ATMOSPHERIC WATER COOLING EQUIPMENT

Figures indicate order of desirability

	Cooling Pond	Spray Pond	Spray Tower	Deck Tower	Mechanical Draft	Indoor Tower
Cost	x	2	1	3	4	5
Area	5	4	3	2	1	x
Height	1	2	3	4–5	4–5	x
Weight per sq ft	x	x	1	3	4	2
Independence of Wind Velocity	6	3	4	5	1–2	1–2
Drift Nuisance	1	6	5	4	2–3	2–3
Make-Up Water Required	1	6	5	4	2–3	2–3
Pumping Head	1	2	3	4–5	4–5	6
Maintenance	2	1	3	4	5	6
Suitability for Congested Districts	x	5	4	3	1	2
Water Quantity Required for Definite Result	6	5	4	1–2	1–2	3

xNot comparable.

tower the upper system may be shut off and a secondary distribution system put in service midway down the height of the tower. The water will be kept above freezing because it will have shorter contact with the air. The mechanical draft tower can be protected by reducing the air flow through the tower, by stopping or reducing the speed of the fans, or by partially closing dampers.

If the system is operated intermittently in freezing weather, water in the basin may freeze and the expansion of the ice may do harm. Freezing during intermittent operation can be prevented only by draining the water basin when it is out of service. On small roof installations, a tank large enough to hold all the water in the system is often installed inside the building and the basin is drained into this by gravity, the pump suction being taken from this inside tank.

A comparison of various types of water cooling equipment is given in Table 4.

Chapter 12

UNIT AIR CONDITIONERS AND COOLERS

Classification of Conditioner Units, Evaporative Cooling, Dehumidifying and Cooling, Make-Up Water Required, Cooling for Summer, Calculation of Cooling Load, Air Temperatures and Volumes, Ratings and Capacities of Units, Location of Units, Commercial Types of Ceiling Units, Floor Units, Ice Units, Portable Units and Accessory Units

AIR conditioning with unit equipment has gained in popularity during the past few years and the types and styles of unit air conditioners vary widely. In selecting or specifying the unit types the results desired should be carefully determined. The essential feature is that such apparatus must *simultaneously* control the temperature, humidity, air motion and distribution within an enclosure. If complete air conditioning functions are not required the simpler and less expensive unit heaters or unit coolers can be used.

Unit air conditioners are devices of small capacity located in the space to be conditioned and may be classified according to design or service. They are not necessarily self-contained but the air treated is not removed from the room being conditioned. Strictly defined, however, the unit air conditioner is self-contained and includes the compressor and condenser.

The unit air conditioner was originally designed to supply proper atmospheric conditions in some room or section of a manufacturing plant where structural conditions or service requirements made installation of a central system un-economic. These units can be readily located in existing buildings or departments and if shifts to meet changing requirements are desirable, they can be moved or the number can be increased to meet growing demands. Because of the flexibility and mobility of the units, tenants of either old or new buildings may have air conditioning even if the building owner does not desire to furnish it. Industrial process conditioning with unit apparatus has had a wide field of application in printing plants, textile mills, candy and tobacco factories, bakeries, drug manufacturing plants, laboratories, meat packing, vegetable and fruit storage rooms.

The application of unit air conditioners for comfort in offices, stores, restaurants, shops, hospitals, hotels and homes developed rapidly and many new designs in which quietness of operation and attention to appearance were emphasized. The unit air conditioner has a distinct field of application and through its use brings the desired temperature, humidity, air circulation and cleanliness to places where a central type plant could not be used.

TYPE OF SERVICE

Because of the variety of arrangements with corresponding variation in the service rendered, unit air conditioners may be classified in three divisions:

A. ALL-YEAR, when supplying:
- (1) Air circulation.
- (2) Air cleaning.
- (3) Air heating in winter.
- (4) Humidification in winter.
- (5) Air cooling in summer.
- (6) Dehumidification in summer.

B. WINTER, when supplying Items (1), (2), (3) and (4).

C. SUMMER, when supplying Items (1), (2), (5) and (6).

The All-Year unit generally takes its air supply from the room and out-of-doors, the cleaning being done by filters or air washers and the circulation being provided by motor-driven blowers or fans. Humidity is obtained usually by evaporation of water or entrained moisture from water sprays or wet surfaces. Dehumidification is accomplished by lowering the temperature below the dew-point and condensing the moisture either by passing the air through water sprays or over cooled surfaces. Heat is supplied to the air by bringing it in contact with steam or water-heated surfaces and in some instances, by the reverse cycle refrigeration method.

Surface cooling may be provided by direct expansion in the surface unit and by circulation of cold water or brine from a central refrigerating plant. The water may be cooled by melting ice or it may be taken from city mains or wells where a low temperature is available. A drip pan or container should discharge to drainage lines in order to remove the condensed moisture or that not taken up by the air.

MAKE-UP WATER REQUIRED

In the ordinary practice of humidifying, the water is constantly recirculated and the only water loss is that which is evaporated into the air and serves to humidify the air. Where air conditioning units recirculate 100 per cent room air and have no outside air connection, the amount of humidifying required for each passage of the air actually is small, as the air in rotating will gradually increase its moisture content to any point desired. If, however, the air is taken from outside, the treatment must increase the humidity to the desired degree (since the air only passes once through the device) and more water will be needed because of the larger requirement for humidifying. It is obvious that the tank must be drained and flushed periodically to dispose of the dirt and dust removed from the air by the water.

The air conditioning units for winter service omit the air cooling and dehumidification functions.

The units for summer use only usually consist of a cooling coil and a fan or fans electrically operated, erected within an enclosure so as to form a single piece of equipment and are intended solely for cooling purposes. A float valve, surge drum, valves and other auxiliary fittings may or may not constitute part of the complete device depending on the type. The

fan draws or forces the air across the cooling surface and discharges the air in selected directions. As a rule the outlets are specially designed so as to deliver the air without the use of ducts and the units are most frequently placed directly in the room to be cooled.

Cooling units are designed to:

1. Circulate the air of the room at a rapid rate.
2. Promote uniformity of temperature throughout the room.
3. Direct the cooled air positively and rapidly to points where it will be most effective.
4. Reduce the amount of space occupied by the cooling equipment.
5. Provide a system readily responding to thermostatic or manual control.
6. Increase the capacity of the cooling surface by passing the air over it at high velocity.
7. Provide, under favorable means, a method of either manual or automatic defrosting of the cooling surface.
8. Provide a neat appearing and sanitary apparatus for obtaining the temperatures and humidities desired.

Cooling units are suitable for operating on gas and liquid refrigerants such as ammonia, methyl chloride, sulphur dioxide, carbon dioxide or on brine or cold water. The choice of refrigerant is governed by the design, material and construction of the unit cooling coil.

CALCULATION OF THE COOLING LOAD

In estimating on the load that the unit air conditioning apparatus must meet a survey should be made of surrounding conditions and the heat losses calculated. The same factors of heat gains and losses are used for all types of comfort cooling.

The sensible heat gains may be summarized as follows:

a. Sunlight load.
b. Radiation load.
c. Infiltration load, depending on air changes.
d. People.
e. Lights.
f. Electrical motors and appliances.
g. Steam and gas appliances.
h. Miscellaneous heat sources.

The latent heat load must be determined separately. This latent heat comes from moisture loss from the drying of air, from people and materials. The heat losses occur through walls, windows, roofs, by infiltration of cold air and humidification. After this total heat has been calculated (see Chapter 8), units are selected from the manufacturers' lists in accordance with their rated capacity. Most units for comfort air conditioning have a cooling capacity ranging from 0.75 tons to 1.25 tons, approximately the amount required for one room of good size or an office, though machines up to 10 ton capacity can be obtained.

Example 1. A fur storage room is to be maintained normally at a temperature of 32 F, but at intervals, the room is to be lowered to 20 F, which temperature will be maintained for several days, after which 32 F will again be held. It is assumed that there is one air change every eight hours. What is the cooling load?

Data: The room is located in the basement of a building that has no outside exposure.

Maximum temperature of rooms surrounding walls and ceilings, 90 F. Floor is on ground. Room has a vestibule entrance. Four lights of 100 watts each will be burning part of the time. No occupants in room except at infrequent intervals. Brine available at a minimum temperature of 10 F.

Size of Room: 25 ft long by 20 ft wide by 9 ft high.
Walls: 6-in. concrete, 4-in. cork and ½-in. plaster finish.
Ceiling: 4-in. concrete, 4-in. cork and ½-in. plaster finish.
Floor: Cinder fill, 6-in. concrete, 4-in. cork, 2-in. concrete.

Solution. The cooling load will be the sum of the following items:

Sun effect	none
Lights: $4 \times 100 \times \frac{3.4}{4}$ (25 per cent use assumed)	340
Cooling unit motor, ⅓ hp	1260
Transmission	
Walls: $90 \times 9 \times (90 - 20) \times 0.066$	3742
Ceiling: $20 \times 25 \times (90 - 20) \times 0.067$	2345
Floor: $20 \times 25 \times (70 - 20) \times 0.065$	1625
Infiltration	
$\frac{4500 \times (90 - 20)}{8 \text{ hr} \times 55.2}$	713
Product Load	
None	
Total	10025
Safety factor (10 per cent)	1002
Total sensible heat load	11027 Btu per hour
Dehumidification Load	
Air at 90 F and 50 per cent rel. hum. 217.6 grs × 50 per cent	108.8
Air at 20 F and 50 per cent rel. hum. 15.01 grs × 50 per cent	7.50
Grains per lb to be removed	101.3
Pounds of air per 8 hr: 4500/14 or 321 lb	
Pounds of air per hr: 321/8 or 40⅛ lb	
Grains per hr: 101.3 × 40⅛ or 4064.6 grs	
Pounds of moisture: 4064.6/7000 or 0.580 lb, 0.58 × 1057 Btu	613
Grand Total	11,640
Tonnage, 11,640/12,000	0.96 tons

Air Temperature and Volume

The drop in temperature of the air through a cooling unit is comparatively small, while the volume handled is large. This feature permits of greater heat transfer per unit of surface and a minimum moisture precipitation. Consequently, the temperature difference between the room and the delivered air is small, making it of little consequence whether the intake of the unit is located at the floor or at the ceiling.

RATINGS AND CAPACITIES

It is standard practice to rate unit air conditioners for heating in Btu per hour and to rate cooling units in Btu per hour at a given dry- and wet-

bulb temperature of air entering the unit with a given refrigerant temperature, maintained within the cooling coil, a certain relationship between sensible and latent heat being designated. In any event the refrigeration unit must have sufficient capacity to care for all of the sensible heat to be absorbed plus the latent heat due to dehumidifying which in some cases may amount to a very substantial part of the load.

At present there is no uniform standard for rating air conditioning units but manufacturers generally have a definite rating for each unit with respect to its air volume or different air volumes, but judgment must be exercised in the selection of equipment so that the component parts are in proper ratio to each other.

LOCATION OF UNITS

The following considerations govern the location of units in a room to be conditioned:

1. Arrangement of product within a space.
2. Location of people within a room.
3. Number and location of units.
4. Air distribution.
5. Convenience of piping and wiring connections.
6. Location of outside air cooling and heating sources.

Most important of these is air distribution and units should be so located as to secure equal distribution to all parts of the room, whether the application is for comfort conditioning or processing. The discharge of cooled air in general should be from a high point and the air traveling back to the inlet of the unit should be at low velocities.

Cold Storage

Air conditioning with unit equipment plays an important role in the cold storage of food. Temperature and humidity control is of prime importance in order to retard bacterial development and retain as much as possible of the original moisture in meats, fish, fruits, vegetables and eggs. Spoilage and shrinkage during long storage periods result in costly losses and may be prevented with proper application and control of air conditions within specified limits. To maintain the high relative humidities generally required, units embodying the features of central plants are considered most suitable. The units are placed to discharge the moist cool air horizontally just below the ceiling so that distribution over the stored product will be uniform and rapid. For economic service 80 per cent relative humidity is the practical upper limit that can be carried. The higher the relative humidity requirement the larger the unit capacity needed.

The control methods and devices used for unit conditioners are discussed in Chapter 14.

TYPES OF EQUIPMENT

Ceiling Units: The types and designs of air conditioning units in production are legion and but a few typical arrangements are shown. New

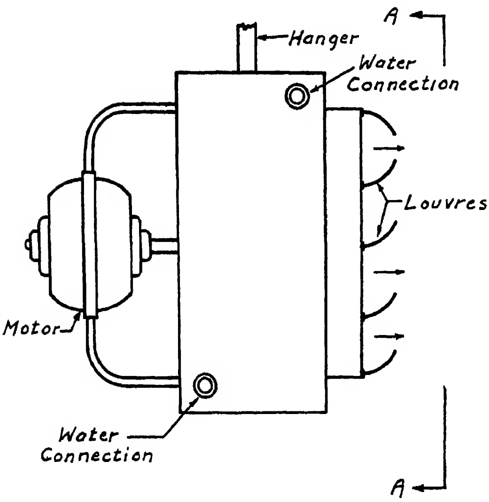


FIG. 1. CEILING UNIT

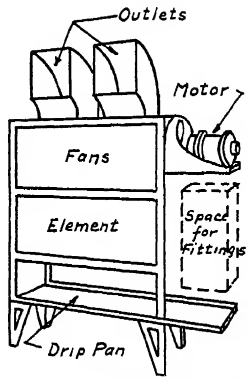


FIG. 4. INDUSTRIAL FLOOR TYPE

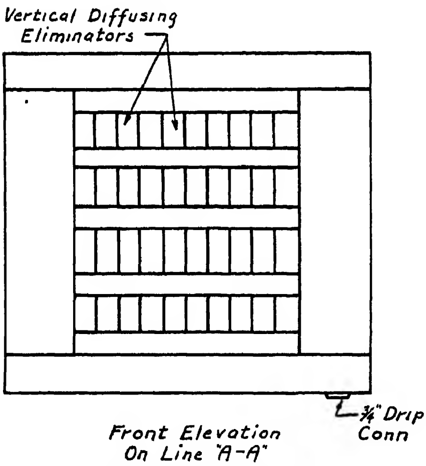


FIG. 2. ELEVATION THROUGH LINE AA

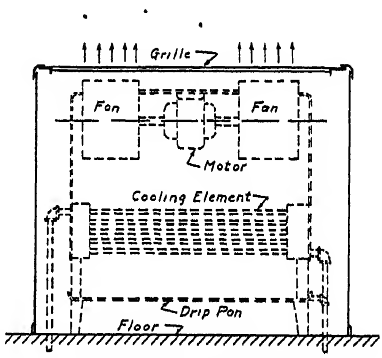


FIG. 5. CABINET TYPE COOLING UNIT

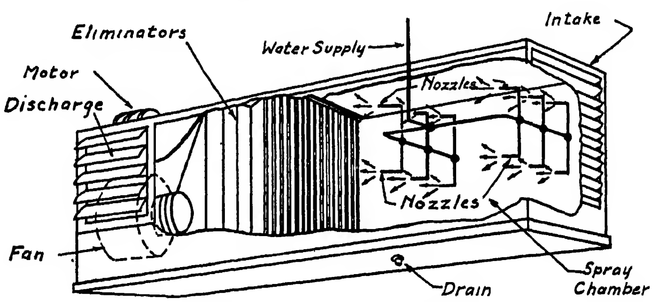


FIG. 3. SUMMER COOLING UNIT

designs are constantly appearing and the general tendency is toward better mechanical construction and more versatile application.

Fig. 1 shows one of the simplest units, where modification of a ceiling unit heater has been made so it can serve as an air conditioner. The assembly shows a motor, propeller fan, extended fin heating and cooling element, drip pan and a double louver arrangement, the vertical louvers being formed by condensate eliminators as illustrated in Fig. 2 which is an elevation of this unit on Line A-A. The whole is enclosed in a metal cabinet and is supported by a ceiling hanger. The horizontal louvers are adjustable so that the air may be directed downward, horizontally, or upward as may be desired. During the winter the unit may be used exactly the same as a unit heater when properly supplied with steam and return connections, while in the summer it may be used as a cooler and dehumidifier if correctly attached to a refrigeration plant or supplied with cooled water of sufficiently low degree. This ceiling type unit is particularly suited for industrial and business applications.

Fig. 3 illustrates a ceiling unit for summer use without any heating provision. This unit, in conjunction with the regular heating system will do a certain amount of humidifying in the winter. It consists of an air washer with the usual water sprays, eliminator plates and air circulating fan. It is hung on the ceiling of the room and takes its air supply from the room through the intake louvers indicated, passes the air through the water spray and eliminators, and then delivers the air back into the room through the discharge outlet provided with louvers of adjustable type. The refrigeration machine may be located at any convenient point and the cooled water circulated to and from the conditioner across the ceiling so that no floor space is lost. This style of unit is for industrial and large office installations and, where the appearance on the ceiling is found to be objectionable, the unit may be placed at some other location (possibly with the refrigeration machine) and the conditioned air may be piped from the unit to the room and back again to the unit through a short system of ducts.

Floor Units: The floor unit for industrial service shown in Fig. 4 has a galvanized iron casing enclosing the heating and cooling element with the fans mounted above and the air discharging from the top through galvanized 90 deg elbows which deliver the air in a horizontal or slightly upward direction. The operating motor for the fan is carried on a bracket at the side and at the bottom a drip pan is provided to catch any condensation that may form, while the space between the pan and the motor bracket may be utilized for the installation of traps and valves. This unit does not attempt to wash or filter the air but when supplied with steam it will heat the air and when supplied with cold water will cool and dehumidify.

A similar arrangement housed in an ornamental type cabinet is shown in Fig. 5. The cooling element, fans and motor, together with the drip pan and piping connections are all housed within a steel casing finished in wood grain. The element is suitable in this case for circulating either hot or refrigerated water but not a refrigerant. This, however, does not apply to all units of this type.

Another floor unit of ornamental type, Fig. 6, has fans below the elements and separate heating and cooling elements. The fan delivers

the air against deflector baffles which spread the air over the element face, and the usual drip pan for precipitation is provided. When this unit is installed for cooling only, the heating element is simply omitted and this arrangement of two elements—one for heating and one for cooling—possesses the advantage of allowing the heating element to be connected to the source of heat with piping entirely separate from the lines used for carrying the refrigerant to the cooling element and without any cross connecting. Thus the unit may be used for warming in the morning and cooling at noon if desired and without the manipulation of special valves. The refrigerant is used direct in the cooling element and units of this type are for service in the home, office or other location where appearances are highly important. The refrigeration machine may be placed at any point where water, drain and electric connections are available.

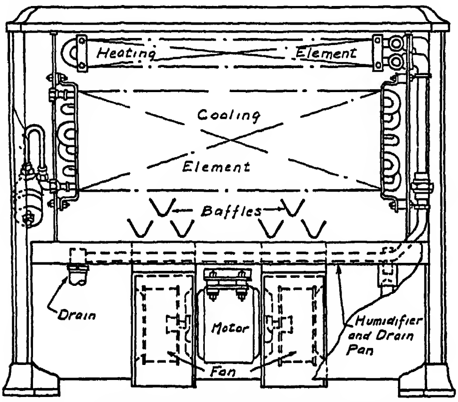


FIG. 6. FLOOR UNIT FOR HEATING AND COOLING

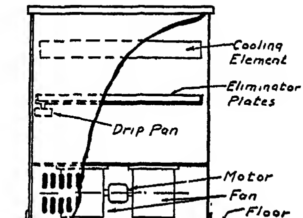


FIG. 7. UNIT FOR COOLING SERVICE

Fig. 7 is a unit for cooling only and consists of two fans operated by a motor, a cooling element and a drip pan all enclosed in a specially insulated cabinet. The element is of copper and may be supplied with cooled water or with any refrigerant which does not attack copper or its derivatives. When occasion demands, a heating element can be added and cross connected to the cooling surface.

A winter and summer conditioning unit, Fig. 8, employs propeller type fans to draw the air in from a horizontal direction, a deflector changes the direction to vertical and the air passes through cooling and heating elements successively before being discharged vertically from the top of the unit. The separate heating and cooling elements make it possible to use this unit for either heating or cooling as desired without alteration of the valving. The refrigerant is in the cooling element, and to prevent waste of cooling water, a solenoid valve closes the flow to the condenser and compressor whenever the fans are stopped.

The unit in Fig. 9 differs from that illustrated in Fig. 5 in that the refrigerant is supplied direct to the cooling element, isobutane being used as the refrigerant in the smaller sizes and methol chloride in the larger units. This unit for the ordinary size office or room requires small space, and operates with the automatic control arrangement indicated. This unit is intended exclusively for cooling.

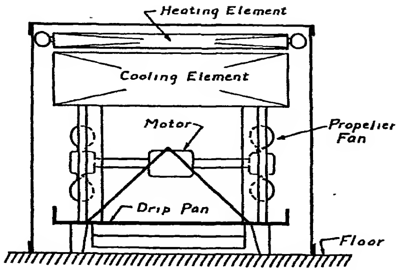


Fig. 8. Winter and Summer Unit

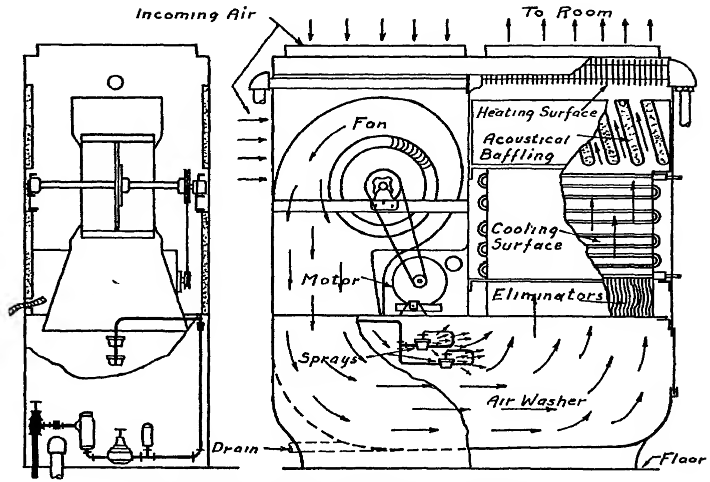


Fig. 10. Unit with Top Inlet and Outlet

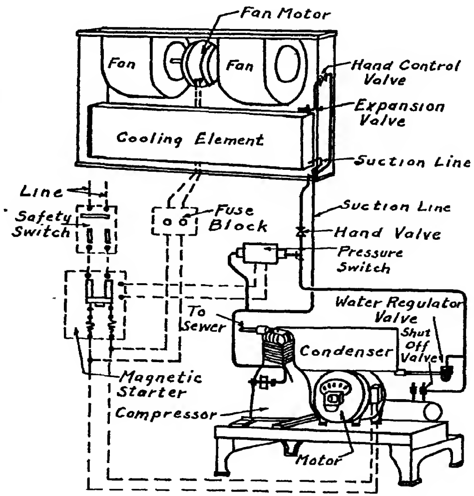


Fig. 9. Room Cooling Unit

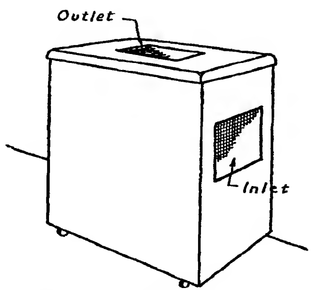


Fig. 12. Accessory Unit for W. A. Furnace

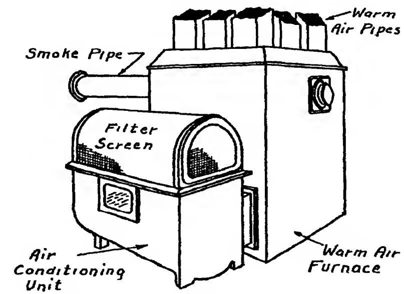
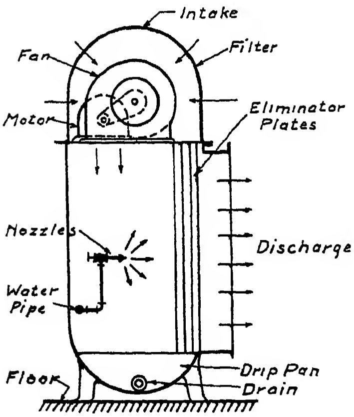
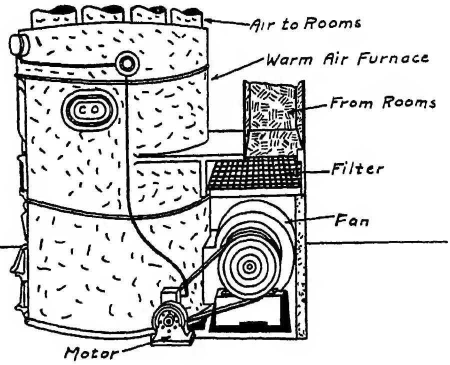
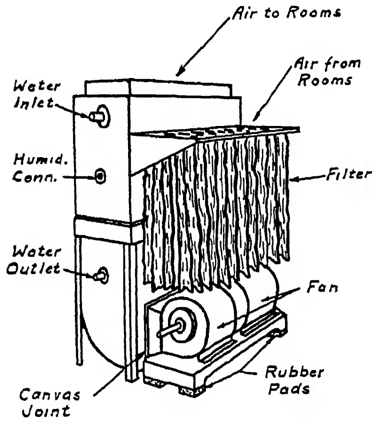
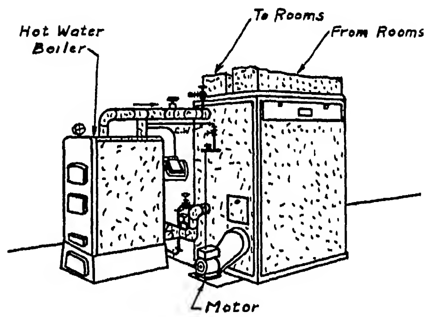


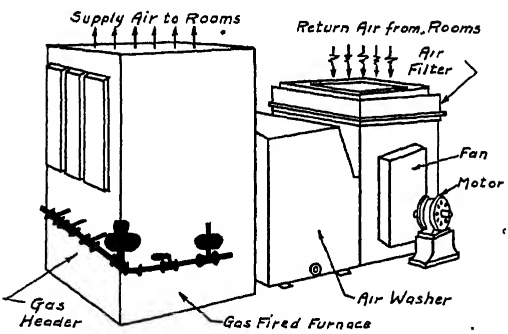
Fig. 11. Portable Ice Unit











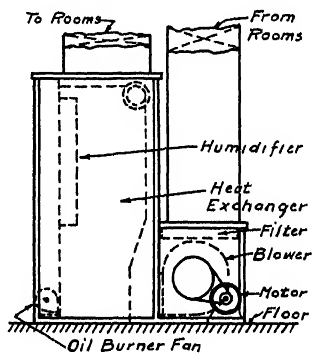


Fig. 13. (top) Cross Section of Furnace Unit
Fig. 15. (center) Heating and Cooling Unit with Cloth Filter
Fig. 17. (bottom) Gas Fired Furnace Unit

Fig. 14. (top) Furnace Accessory Unit
Fig. 16. (center) Unit with Hot Water Boiler
Fig. 18. (bottom) Oil Fired Unit

A radical departure in arrangement is shown in Fig. 10 where both the air inlet and exit are at the top of the unit. The fan is located in the top portion at one side and discharges the air toward the bottom where it turns and passes horizontally through an air washer equipped with atomizing sprays and then turns so as to pass vertically upward through eliminators, cooling surface and heating surface before leaving the unit. With steam or hot water connected to the heating surface, tempered water to the sprays and refrigerated water to the cooling element this unit gives controlled temperature, humidity, air cleaning and air movement in the summer and winter. Air washing is continued in the summer to remove room odors. Acoustical treatment of the housing and the outlet baffles permits installations where the noise requirements are exacting.

Ice Units: In commercial type units using ice there is little difference between the outward appearance and that of the mechanical units. The ice using units are enclosed in the same metal housings, are operated by similar fan arrangements and employ an ice container as cooling surface. Fig. 11 illustrates one type of portable machine. The capacity is limited and, ordinarily, it must not be expected that a single ice-cooled unit will give the same cooling effect as a single room unit of mechanical design. In hotels or hospitals where the unit may be desired in different rooms from day to day and where elevator service is available so that the units may be taken into the utility rooms for emptying and re-charging there seems to be a real field of application.

Portable Units: A portable type of mechanical unit is now available which is suitable for cooling and dehumidifying only, this unit being equipped with rubber tired wheels, so that it may be shifted with ease from room to room and with a rubber hose 15 ft long for connecting to a nearby water faucet and with an electric extension cord and plug.

Accessory Units: Under this classification may be included every type of unit which has been developed as an accessory to an existing or proposed system of warm air distribution primarily designed for heating service. Some of these accessory units for warm air furnace systems simply provide a fan and air filter, while others include humidifying and cooling apparatus. The performance of this equipment is influenced by the following factors and conservative manufacturers claim only a moderate degree of added comfort:

1. The outside temperature and humidity conditions.
2. The heating system to which the device is attached.
3. The location of the outlets on the heating system.
4. The conditions surrounding the house or apartment such as construction, exposure to sun.

The introduction of cooler night air into sleeping quarters or living rooms will give greater comfort on summer nights and for some time during the following day. The added humidity during the winter will do much to improve indoor conditions but where specified conditions must be obtained and held, a special type of installation must be used. (See Chapter 2).

Fig. 12 illustrates an accessory unit for a warm air furnace installation this unit consisting of a large semi-cylindrical screen on the top of fine copper mesh through which the air is drawn by a fan blowing down into a

spray chamber where the air is washed and humidified and then passes through eliminators into the lower portion of the furnace casing. (See cross section Fig. 13). If desired a cooling element and mechanical refrigeration may be added for cooling in summer.

In Fig. 14 return air from the rooms is drawn by a fan through a dry mat type air filter and is then delivered into the bottom of the furnace casing. While this unit makes no attempt to utilize cooling water or refrigeration, humidification during the winter is obtained from humidifiers located in the furnace.

A more elaborate unit, Fig. 15, for heating and cooling employs a cloth filter of bag design and a double element for heating and cooling. In summer, water is run through both elements either direct from the street main or from some mechanical cooling equipment, while in the winter hot water is supplied from a hot water boiler using the same surface.

These units are provided with a single element in order to reduce first cost and, when so equipped, will do everything that the double element unit will accomplish except that less cooling is available in the summer. Humidity is provided by spraying water between the two elements. Noise is reduced by the use of rubber pads under the bedplate supporting the fans and by the connecting of the fans to the metal work of the discharge through canvas connections. Fig. 15 shows the outer casing removed.

Another design using hot water heating boilers is given in Fig. 16 where the unit takes air from both inside and outside, utilizing hot water from the boiler for winter heating and water, either iced or mechanically cooled, for summer cooling. The boiler may be coal, oil or gas fired and the air is humidified or dehumidified as the conditions require; the same element is utilized for heating and cooling, the piping being cross connected.

Where gas can be used, Fig. 17 shows a unit consisting of an air filter, motor-driven fan, air washer and gas-fired steel furnace which warms the air during the winter season. No refrigeration is used with this equipment, the idea being that the air washer during the summer will lower the air temperature sufficiently when the humidity is low while at night, the apparatus is run without the washer when the air is cooler but of relatively higher humidity.

For oil fuel the unit shown in Fig. 18 can be installed to obtain filtered, warmed and humidified air. An oil burner and a heat exchanger provide the heat. A cooling section may be inserted between the fan and the heat exchanger, cold water being circulated through the cooling element. This equipment is completely automatic and is thermostatically controlled. A room thermostat starts the oil burner whenever the temperature falls and the rising temperature in the heat exchanger causes a second thermostat to start the fan. As soon as the temperature in the house rises to normal the room thermostat shuts down the oil burner which in turn operates the thermostat controlling the fan.

Chapter 13

UNIT HEATERS AND VENTILATORS

Types of Unit Heaters, Heating Media, Entering and Delivery Temperature, Output of Unit Heaters, Direction of Discharge, Boiler Capacity, Direct-Fired Units, Unit Ventilators, Split and Combined Systems, Location of Unit Ventilators, Capacities

A UNIT heater consists of a combination of a heating unit and a fan or blower having a common enclosure, and placed within or adjacent to the space to be heated. Generally, no ducts are attached to the inlets or outlets. A unit ventilator is similar in principle of operation to a unit heater, but is designed to use all or part out-door air with or without alternate provision for handling recirculated air. Unit heaters are designed mainly for factory and industrial use, whereas unit ventilators are intended largely for school and office ventilation and heating.

Unit heaters are designed to:

1. Circulate the air in the building at a rapid rate.
2. Promote uniformity of temperature and quick heating-up.
3. Direct the heated air so as to accomplish the positive and rapid placing of the heat where it is effective.
4. Reduce the temperature differential between floor and ceiling.
5. Reduce the number of heat emitting units and simplify piping and installation.
6. Increase the capacity of the heating surface by passing the air over it at high velocity.
7. Provide a system by which room temperatures are readily controlled manually or by thermostats.

TYPES OF UNIT HEATERS

There are many types of unit heaters available. Most of them employ heating coils to be supplied with steam or hot water. Some are mounted on the floor, whereas others are designed for suspension overhead. Heating surfaces in the form of pipe coils, non-ferrous tubes or shapes with extended surfaces, cast-iron, and pressed and built-up sections of the cartridge or automotive type are all used in unit heater construction.

Among the unit heaters available are types having from one to four outlets per heater which may be arranged to discharge in selected directions and which will project their heating effect over distances of from 30 to 200 ft from the heater, depending upon the capacity of the heater and upon the design of the fans and outlets. These heaters have been successful when placed as far as 400 ft from each other. This makes it possible to select the heater location best suited to the production layout in factories. There are available propeller fan type heaters of smaller

Table 1. Constants for Determining the Capacity of Blow-Through Type Unit Heaters for Various Steam Pressures and Temperatures of Entering Air

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Entering Air											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.538	1.446	1.369	1.273	1.191	1.110	1.034	0.956	0.881	0.809	0.739	0.671
2	1.585	1.495	1.405	1.320	1.237	1.155	1.078	**1.000**	0.926	0.853	0.782	0.713
5	1.640	1.550	1.456	1.370	1.289	1.206	1.127	1.050	0.974	0.901	0.829	0.760
10	1.730	1.639	1.545	1.460	1.375	1.290	1.211	1.131	1.056	0.982	0.908	0.838
15	1.799	1.708	1.614	1.525	1.441	1.335	1.275	1.194	1.117	1.043	0.970	0.897
20	1.861	1.769	1.675	1.584	1.498	1.416	1.333	1.251	1.174	1.097	1.024	0.952
30	1.966	1.871	1.775	1.684	1.597	1.509	1.429	1.346	1.266	1.190	1.115	1.042
40	2.058	1.959	1.862	1.771	1.683	1.596	1.511	1.430	1.349	1.270	1.194	1.119
50	2.134	2.035	1.936	1.845	1.755	1.666	1.582	1.498	1.416	1.338	1.262	1.187
60	2.196	2.094	1.997	1.902	1.811	1.725	1.640	1.555	1.472	1.393	1.314	1.239
70	2.256	2.157	2.057	1.961	1.872	1.782	1.696	1.610	1.527	1.447	1.368	1.293
75	2.283	2.183	2.085	1.990	1.896	1.808	1.721	1.635	1.552	1.472	1.392	1.316
80	2.312	2.211	2.112	2.015	1.925	1.836	1.748	1.660	1.577	1.497	1.418	1.342
90	2.361	2.258	2.159	2.063	1.968	1.880	1.792	1.705	1.621	1.541	1.461	1.383
100	2.409	2.307	2.204	2.108	2.015	1.927	1.836	1.749	1.663	1.581	1.502	1.424

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

capacity with outlet velocities of from 300 to 800 fpm, and these may be placed from 30 to 100 ft apart.

HEATING MEDIA

Unit heaters are made to operate with hot water, or with steam at high or low pressure. When high pressure steam is used, the heater must be especially designed for that purpose.

The present day tendency is to use steam at low pressure when the boiler supplies steam for heating purposes only, and when the transmission lines are short. Many industrial plants, however, generate steam at high pressure either for long distance transmission or for process uses. When such high pressure steam is available in sufficient quantity for the peak load, it is economical usually to use high pressure unit heaters. Thus the line pressure may be turned into the units directly without reducing valves and the condensation may be trapped to overhead returns when desired. Smaller coils may be used in high pressure heaters to lessen their cost and to produce a final temperature which will not be too high.

ESTIMATING HEAT LOSSES

The heat losses of a building to be equipped with unit heaters are determined in the same manner as for any other heating system, excepting so far as the unit heaters may change the air temperature at the ceiling or at the mean height of the walls. (See Chapter 7).

Unit heaters may be arranged to recirculate the air or to supply warmed air from the outside for ventilation or to make up air exhausted.

If all or a part of the air is to be taken in from out-of-doors, the heat necessary to warm this air from the outside temperature to the inside temperature must be added to the transmission or other losses. Unit heaters of the number and size needed to furnish the total heat required are then selected from the manufacturers' rating tables, using these ratings at the steam pressure to be used and at the temperature at which the air will enter the heater.

ENTERING AND DELIVERY TEMPERATURES

For recirculating heaters with intakes at the floor level, the temperature to be maintained in the room should be used as the temperature of the air entering the heater. Where suspended heaters are used without any intake boxes extending down to the floor level, a higher entering air temperature should be used than that at which the room is to be maintained. With suspended heaters taking in air at some distance above the floor, the temperature variation from floor to ceiling may reach as much as 1½ deg for each foot of elevation during periods when the maximum capacity of the heaters is required. Unit heaters taking in recirculated air at the floor level should maintain temperature differentials of less than 1 deg per foot of elevation when the maximum capacity of the heaters is required. These temperature differences per foot of elevation are less than the corresponding variations per foot of elevation for spaces heated by direct radiation.

TABLE 2. CONSTANTS FOR DETERMINING THE CAPACITY OF DRAW-THROUGH TYPE UNIT HEATERS FOR VARIOUS STEAM PRESSURES AND TEMPERATURES OF ENTERING AIR

(Based on Steam Pressure of 2-lb Gage and Entering Air Temperature of 60 F)

Steam Pressure Lb per Sq In.	Temperature of Air Entering Heater											
	−10°	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
0	1.483	1.405	1.329	1.253	1.178	1.105	1.032	0.962	0.892	0.822	0.754	0.688
2	1.520	1.442	1.363	1.290	1.215	1.141	1.069	**1.000**	0.930	0.861	0.792	0.728
5	1.565	1.485	1.410	1.334	1.260	1.187	1.114	1.045	0.975	0.906	0.838	0.771
10	1.637	1.558	1.480	1.403	1.328	1.253	1.182	1.112	1.042	0.973	0.903	0.838
15	1.688	1.610	1.533	1.458	1.382	1.310	1.239	1.168	1.099	1.028	0.960	0.895
20	1.728	1.649	1.572	1.498	1.421	1.350	1.278	1.208	1.138	1.070	1.002	0.936
30	1.803	1.725	1.648	1.572	1.497	1.423	1.352	1.281	1.212	1.145	1.078	1.010
40	1.864	1.787	1.710	1.637	1.563	1.491	1.420	1.350	1.282	1.215	1.148	1.081
50	1.927	1.850	1.773	1.700	1.628	1.554	1.483	1.416	1.347	1.278	1.211	1.145
60	1.973	1.897	1.820	1.748	1.673	1.601	1.531	1.463	1.394	1.325	1.260	1.194
70	2.018	1.943	1.869	1.795	1.722	1.651	1.582	1.512	1.443	1.377	1.310	1.243
75	2.043	1.970	1.895	1.822	1.750	1.680	1.609	1.540	1.471	1.402	1.333	1.268
80	2.064	1.988	1.914	1.841	1.770	1.698	1.629	1.560	1.491	1.422	1.354	1.288
90	2.102	2.028	1.951	1.878	1.804	1.732	1.661	1.590	1.523	1.457	1.387	1.321
100	2.150	2.071	1.994	1.919	1.845	1.770	1.700	1.630	1.560	1.492	1.425	1.359

Note.—To determine capacity at any steam pressure and entering temperature, multiply constant from table by rated capacity at 60 F entering and 2 lb pressure.

A rapid recirculation or turnover of the air in the room will give fuel economy. This requires the selection of heaters having a liberal air capacity for the required heat output, which in turn means a relatively low final temperature. Extremely low final temperatures can be had only at the expense of larger heaters and increased power, so that an economic limit is imposed. Contributing conditions vary too widely to permit of a suggested standard, but in general for heating purposes it is advisable to use a delivery temperature not more than 70 deg above the average room temperature desired.

OUTPUT OF UNIT HEATERS

It is standard practice to rate unit heaters in Btu per hour at a given temperature of air entering the heater and at a given steam pressure maintained in the coil. Steam at 2 lb pressure and air entering at 60 F are used as the standard basis of rating[1]. The capacity of a heater increases as the steam pressure increases, and decreases as the entering air temperature increases. The heat capacity for any condition of steam pressure and entering air temperature may be calculated approximately from any given rating by the use of factors in Tables 1 and 2. Table 1 is for blow-through and Table 2 is for draw-through unit heaters. These tables are accurate within 5 per cent.

The ratings customarily published for unit heaters apply only for recirculation and free discharge, unless otherwise noted in the rating tables. If outside air intakes, filters or ducts on the discharge side are used with the heater, proper consideration should be given to the reduction in air and heat capacity that will result because of this added resistance.

The percentage of this reduction in capacity will depend upon the characteristics of the heater and on the type, design, and speed of the fans employed, so that no specific percentage of reduction can be assigned for all heaters for a given added resistance. In general, however, disc or propeller fan units will have a larger reduction in capacity than housed fan units for a given added resistance, and a given heater will have a larger reduction in capacity as the fan speed is lowered. When confronted with this problem the ratings under the conditions expected should be secured from the manufacturer.

When steam supplied to the heaters contains superheat, the capacity of the heater will be but slightly less than with saturated steam at the same pressure. Recent tests indicate that the reduction of capacity from this cause is negligible for superheat up to 50 deg and will not exceed 3½ per cent for any degree of superheat.

Heaters may be distributed through the central portions of a room discharging toward exposed surfaces, or may be spaced around the walls, discharging along the walls and inward as well, when there are considerable roof losses.

In general, it is better to direct the discharge from the unit heaters in such fashion that rotational circulation of the entire room content is

[1]See A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Heaters. (A.S.H.V.E. Transactions, Vol. 36, 1930).

set up by the system rather than to have the heaters discharge at random and in counter directions.

DIRECTION OF DISCHARGE

Various types and makes of unit heaters are illustrated in the Catalog Section of The Guide. Usually hot blasts of air in working zones are objectionable, so heaters mounted on the floor should have their discharge

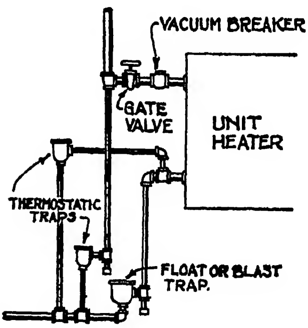


Fig. 1. Unit Heater Connections Where Condensation Is Returned to Vacuum Pump

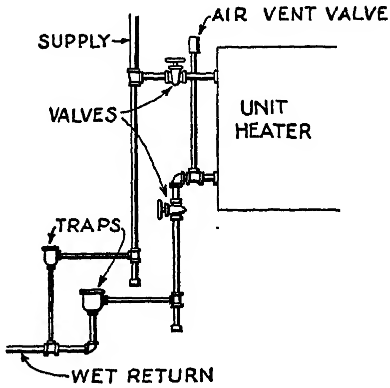


Fig. 2. Unit Heater Connections Where Condensation Is Returned to Boiler Through Wet Return

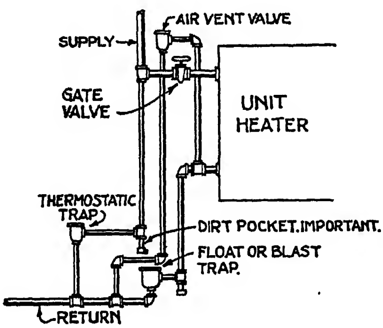


Fig. 3. Unit Heater Connections Where Condensation Is Returned to Condensation Pump or Hot Well

outlets above the head line and suspended heaters should be placed in such manner and turned in such direction that the heated air stream will not be objectionable in the working zone. In the interest of economy, however, the elevation of the heater outlet and the direction of discharge should be so arranged that the heated air shall be brought as close to the head line as possible, yet not into the working zone. In general, the higher the elevation of the unit, the greater the volume and velocity required to bring the warm air down to the working zone, and consequently, the lower the required temperature of the air leaving the unit.

BOILER CAPACITY

The capacity of the boiler should be based on the rated capacity of the heaters at the lowest entering air temperature that will occur, plus an allowance for line losses. Ordinarily for recirculating heaters the lowest entering temperature will occur at the beginning of the heating period and is usually taken as 40 F, while for heaters taking air from outdoors the lowest entering temperature will be the extreme outdoor temperature expected in the district. No greater allowance in boiler capacity beyond the calculated heat demand need be added in order to supply unit heaters than for any other type of system.

It is unwise to install a single unit heater as the sole load on any boiler, particularly if the unit heater motor is started and stopped by thermostatic control. The wide and sudden fluctuations of load that occur under such conditions would require closer attendance to the boiler than is usually possible in a small installation. Where oil or gas is used to fire the boiler, it is possible by means of a pressurestat to control the boiler, in response to this rapid fluctuation. In most cases, however, and particularly where the boiler is coal-fired, it is advisable to use two or more smaller heating units instead of one large unit.

Steam pressures below 5 lb can be used with safety for recirculating unit heaters when their coils are designed for the purpose and when proper provision is made for returning the condensate. If heaters are to take in air that may be at a temperature below freezing, however, a steam pressure of not less than 5 lb should be maintained on the heater coils.

QUIETNESS OF OPERATION

In selecting unit heaters, attention should be given to the degree of quietness required for the installation.

No given fan speed may be applied as a measure of relative quietness to fans of different designs and proportions. Quietness is a function of type, diameter, blade form and other variables besides speed, and all these must be taken into account. In general small fans may be run at higher motor speeds than large fans with equal quietness.

UNIT HEATER CONNECTIONS

Piping connections for unit heaters are similar to those for other types of fan-blast heaters. Typical connections are shown in Figs. 1, 2 and 3.

One-pipe gravity and vapor systems are not recommended for unit heater work.

For two-pipe gravity, or pump and receiver systems the return from each unit should be fitted with a heavy-duty or blast trap and an air valve should be connected into the return header of each unit. Pressure-drop must be compensated for by elevation of the heater above the water line of the boiler or of the receiver.

On vacuum systems the return from each unit should be fitted with a large capacity trap to discharge the water of condensation and with a thermostatic air valve for eliminating the air, or with a heavy-duty trap

for handling both the condensation and the air, provided the air finally can be eliminated at some other point in the return system.

For high pressure systems the same kind of traps may be used as with vacuum systems, except that they must be constructed for the pressure used. If the air is to be eliminated at the return header of the unit, a high pressure air valve can be used; otherwise the air may be passed with the condensate through the high-pressure return trap.

The connections for steam and return piping to unit heaters must always be calculated on the basis of the high heat emission or condensation rate of such devices, usually by reducing the heat to be supplied to square feet of equivalent radiation by dividing by 240, and then using the pipe-size tables given in Chapter 32.

ALL-ELECTRIC UNIT HEATERS

The foregoing discussion relates generally to units in which steam, vapor, or hot water are used as the heating medium. On rare occasions electrical resistances are used as the heating element. These are applied only where electric power is abundant and cheap and where other forms of fuel are scarce and expensive. (See Chapter 38).

DIRECT-FIRED UNITS

A recent development in gas burning equipment is the direct-fired industrial unit heater. These heaters are of the warm air type and are equipped either with fans or with blowers which cause the air to pass over the heating surfaces at a fairly high velocity and then direct the warm air in to the space to be heated in such a way as to give a positive distribution of the heated air. As is the case with the steam fed unit heaters, the gas fired appliances may be used for heating stores, shops, warehouses, etc. They usually are suspended in the space to be heated and in most instances leave the entire floor and wall area free for commercial use. Partial or complete automatic control also may be secured on appliances of this type. This type of heater is often used for temporary heat during building construction or where the installation of a steam or hot water plant is for some reason not justified.

TURBINE-DRIVEN HEATERS

Where high pressure steam is available it is sometimes used to drive a steam turbine direct-connected to the unit heater. The exhaust from this turbine, reduced in pressure, is then passed into the heating coil where it is condensed and returned to the boiler.

INDUSTRIAL USES

In addition to their prime function of heating buildings, unit heaters may be adapted to a number of industrial processes, such as drying and curing, with which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption, such as fog removal in dye-houses, or for the prevention of condensation on ceilings or other cold surfaces of buildings in

which process moisture is given off. When such conditions are severe, it is necessary that the heaters draw air from outside in enough volume to provide a rapid air change and that they operate in conjunction with ventilators or fans for exhausting the moisture-laden air. (See discussion of condensation in Chapter 7).

Information on the control of unit heaters will be found in Chapter 14.

UNIT VENTILATORS[2]

A unit ventilator must be pleasing in design because it is generally used where it must harmonize with the furniture or with the decorative scheme. It consists usually of a rectangular steel cabinet finished with an enameled surface and containing the following necessary or optional parts:

1. Outside air inlet.
2. Inlet damper for closing the opening to the outside air inlet when the unit is not in use.
3. Adhesive or dry type filters for cleaning the air (optional).
4. A heating element usually of special design and intended for low pressure steam.
5. Motor and fan assembly.
6. Mixing chamber where warm and cold air streams are brought together. (No mixing chamber is normally provided where sectional type heating units are used).
7. Outdoor air inlet and recirculating air mixing damper (optional).
8. Humidifying arrangement[3] (optional).
9. Device for ozonizing air (optional).
10. Discharge grille or diffuser.
11. Temperature control arrangement.

The primary functions of a unit ventilator are:

1. To supply a given quantity of outdoor air for ventilation or to mix indoor and outdoor air.
2. To warm the air to approximately the room temperature if the unit is intended for ventilation only, or to a higher temperature if it is intended to take care of all or a part of the heat transmission losses from the room.
3. To control the temperature of the air delivered so as to prevent both cold drafts and overheating.
4. To deliver air to the room in such a manner that proper distribution is obtained without drafts.
5. To recirculate room air for the purpose of heating when ventilation is unnecessary, or to partly recirculate it, mixing inside and outside air.
6. To perform all its functions without objectionable noise.

In addition to these functions, unit ventilators frequently are arranged so that the air supplied may be cleaned by means of filters of either the dry or viscous type. If filters are used, the proper allowance must be made for the increased resistance offered to the air flow. Humidifiers in unit ventilators are optional.

1. *Air Supply for Ventilation.* The outdoor air supply for ventilation is brought about by motor-driven fan or fans operated at comparatively

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[2]A roof ventilator is sometimes termed a *unit ventilator*. For information on roof ventilators, see Chapter 4.

[3]If the unit includes provision for control of humidity, it is usually called a *unit conditioner*. See Chapter 12.

low speeds and located in the lower part of the cabinet, the back of the cabinet being connected to the outside through rust-proof louvers and screens. Air quantities may be estimated on the basis of data given in Chapter 2. (See A.S.H.V.E. Ventilation Standards).

2. *Warming Incoming Air.* The air is heated by blowing it through specially designed extended heating surfaces. The amount of heating surface to be provided in the unit is of course determined by the volume of air to be heated and the temperature range. If the unit is to be used for supplying air for ventilation only, the heating surface must be sufficient to maintain a final air temperature of about 70 F. If the unit is to be used for heating as well as for ventilation, the heating surface must be sufficient to maintain the necessary final air temperature for the conditions involved.

3. *Control of Temperature.* This is accomplished by controlling the temperature of the air discharged from the unit in one of two ways: *first*, by the automatic operation of a mixing damper which controls the relative quantities of air being blown through the heating unit or by-passed around it; and *second*, by dividing the heating unit into two or more sections and controlling (*i.e.*, throttling) the steam so that the proper amount of steam will be supplied to as many sections as required.

The outside air inlet damper and recirculating damper (where one is provided) should be so connected that there will be an uninterrupted supply of air to the fans at all times the unit is in operation. These dampers may be operated by hand or by pneumatic or electric motors controlled from some central point such as the engineer's office, or automatically at the unit itself. Provision should be made for the inlet damper to close automatically whenever the fans are shut down. The temperature of the air from the machine should be regulated automatically by thermostats in the room which control the position of the by-pass dampers below the heating elements, or the number of sections to which steam is supplied. In addition to the room thermostat, a thermostat is frequently provided in the air stream from the unit to maintain a minimum air stream temperature. Thermostats for controlling by-pass dampers must be of the intermediate type to hold the dampers in intermediate positions to prevent objectionable drafts. When direct radiators are used in conjunction with unit ventilators, the control is usually arranged so as automatically to open the valves to the direct radiators when the room temperature falls about 2 deg below the setting of the thermostat for the unit ventilator. Another arrangement opens the radiator valve whenever the unit ventilator control reaches the full heating position. Further information on this subject is contained in Chapter 14.

4. *Distribution.* This function is governed by the proper selection and location of the unit. Diffusion and distribution are dependent upon a relatively high velocity air stream discharged in a generally vertical direction, and in order to insure satisfactory diffusion in the room the final temperature of the air discharged from the unit must be kept as low as possible. With a final temperature above 110 F, excessive stratification of the air may be experienced. Troublesome drafts may be eliminated to a large extent if a static pressure is built up in the room, which can be done only if the vent from the room is small.

5. *Recirculation* reduces fuel consumption and aids in heating up rooms. Certain units are designed to recirculate all air at all times, except when the admission of outside air is needed to regulate room temperatures. Under this arrangement, the outside air for ventilating purposes is obtained solely from the infiltration, but the amount thus obtained is ordinarily insufficient to meet legal ventilating requirements. Recirculation is therefore prohibited by ordinance in some communities.

6. *Quiet Operation.* The unit ventilator is generally set close to the occupants of the room where activities are being carried on which preclude the use of noisy equipment. Quiet operation is therefore imperative.

SPLIT AND COMBINED SYSTEMS

In the *split* system the unit is used primarily for ventilation, delivering the air to the room at very near the room temperature, sufficient separate direct heaters being placed in the room to heat it to the desired temperature, independently of the unit. With the split system the separate direct heaters tend to improve the air circulation and diffusion within the room, since they constantly pull the cold air from the floor, warm it and use it to counteract the window chill.

Where the unit ventilator selected may have a capacity more than sufficient to warm the air needed to meet the ventilating requirements, a corresponding reduction may be made in the amount of direct heating surface installed. The greater the amount of excess capacity of the unit, the more efficient will be the temperature regulation of the room. The split system permits the heating of the room during failure of electric current, since the direct radiators will furnish heat, but it permits a careless operator to avoid operating the ventilating equipment.

A combined system employs the unit ventilator alone, its capacity being sufficient both for ventilation and for supplying the heat loss. Direct heating surface is omitted altogether. It becomes necessary then that the fan be running whenever the room is to be heated and this also gives assurance of ventilation. The cost of installation of a combined system is usually less than that of a split system and there is less danger of overheating.

Central fan systems (Chapters 9 and 22) as well as unit ventilator systems may be designed for split or combined operation.

LOCATION OF UNIT VENTILATOR

The location of the unit ventilator in a room is important. Wherever possible it should be placed centrally on an outside wall. It is difficult to obtain proper air distribution if the unit is erected either on an inside wall or in a corner of the room. Standard units discharge the air stream upward, but for special cases units may be installed to discharge air horizontally. Units may be set away from the wall or partially recessed into the wall to save space without materially affecting the results. The air inlet may enter the cabinet at the back at any point from top to bottom; all units, however, should be arranged to operate with free air inlet and free discharge.

SIZE AND LOCATION OF VENT

The size and location of the vent outlet is important. In many cases the sizes for public buildings are regulated by law, but the location of the vents generally is left to the discretion of the engineer.

Best results have been obtained with a velocity through the vent openings nearly equal to that at which the air is introduced into the room, thus maintaining a slight pressure in the room. Calculated velocities at the vent openings of from 600 to 800 fpm produce the best diffusion results from this system.

The cross-sectional area of the vent flue itself may be figured on the basis of 15 sq in. of flue for each 100 cfm. Thus the vent flue area of a flue for a room equipped with one 1200 cfm unit ventilating machine would be 180 sq in. The area of vent flue opening from the room may be figured on the basis of 25 sq in. per 100 cfm.

In buildings provided with wardrobes or cloakrooms the vents may be so located that the air shall pass through these spaces, heating and ventilating them with air which otherwise would be passed to the outside without being used to the best advantage. Many state codes for ventilation of public buildings make this arrangement mandatory.

The individual vent flue and the corridor or central vent flue are the types generally used with unit ventilators. The individual vent flue system provides for separate vent flues from each room. The corridor vent or central vent outlet system provides for outlets centrally located to take care of a group of classrooms, the air passing from the classrooms, wardrobe or cloakroom to the corridor, and from the corridor to the central outlets. The grilles between the wardrobes and corridors may be near the ceiling, provided the air leaves the classroom to enter the wardrobes near the floor in the classrooms. A velocity of 800 to 900 fpm may be used for the central outlet from the corridor.

Where the law and the type of building construction will permit, the escaping spent air from the building may circulate throughout the attic on its way out.

CAPACITIES

Unit ventilators are available in air capacities ranging from 450 cfm to 5000 cfm and with corresponding heat capacities (above that required for ventilation purposes based upon an outside temperature of zero and an inside temperature of 70 F) ranging from 125 to 600 sq ft of equivalent direct heating surface. Some manufacturers furnish a unit with several heating capacities for each air capacity, thus enabling the engineer to select the unit best adapted to the heating and ventilating load. Capacities should be determined in accordance with the A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators[4]. Typical capacities are given in Table 3.

The amount of heat to be supplied by the unit ventilator will depend on the amount of air passed through the unit and the temperature range through which the air is heated. The weight of air (W) to be circulated per hour is fixed by the ventilating requirements.

[4]Adopted 1932. See A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

If no direct heating surface (radiation) is installed, the combined heating and ventilating requirements must be taken care of by the unit ventilators, and the total heat to be supplied is obtained by means of the following formulæ:

$$H_t = 0.24\,W\,(t_y - t_o) \tag{1}$$

$$W = dQ \tag{2}$$

$$t_y = \frac{H}{0.24W} + t \tag{3}$$

where

- d = density of air, pounds per cubic foot.
- H = heat loss of room, Btu per hour.
- H_v = heat required to warm air for ventilation, Btu per hour.
- H_t = total heat requirements for both heating and ventilation, Btu per hour $= H + H_v$.
- Q = volume of air introduced for the ventilating requirements, cubic feet per hour.
- t = temperature to be maintained in the room.
- t_o = outside temperature.
- t_y = temperature of the air leaving the unit.
- W = weight of air circulated, pounds per hour.
- 0.24 = specific heat of air at constant pressure.

From Equations 1, 2 and 3:

$$H_t = H + 0.24\,dQ\,(t - t_o) \tag{4}$$

Example 1. The heat loss of a certain room is 24,000 Btu per hour, and the ventilating requirements are 1000 cfm. If the room temperature is to be 70 F and all air taken from the outside is zero, what will be the total heat demand on the unit if it is required to provide for both the heating and ventilating requirements (combined system).

Solution. $H = 24{,}000$; $d = 0.075$
$Q = 1000 \times 60 = 60{,}000$ cfh; $t = 70$ F; $t_o = 0$.

Substituting in Equation 4:

$$H_t = 24{,}000 + 0.24 \times 0.075 \times 60{,}000\,(70 - 0) = 99{,}600 \text{ Btu}$$

$$t_y = \frac{24{,}000}{0.24 \times 0.075 \times 60{,}000} + 70 = 92.2 \text{ F}$$

TABLE 3. TYPICAL CAPACITIES OF UNIT VENTILATORS FOR AN ENTERING AIR TEMPERATURE OF ZERO.

CUBIC FEET OF AIR PER MINUTE	TOTAL CAPACITY IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)	CAPACITY AVAILABLE FOR HEATING THE ROOM IN SQUARE FEET OF EQUIVALENT DIRECT HEATING SURFACE (RADIATION)	FINAL AIR TEMPERATURE (DEG FAHR)
600	285	95	105
750	350	115	105
1000	455	150	105
1200	565	190	105
1500	705	235	105

If all of the air is recirculated, the total heat required is of course the same as the heat loss of the room, or

$$H_t = H = 0.24\ W\ (t_y - t) \tag{5}$$

If the heat loss of the room is to be taken care of by the direct heating surface, the unit ventilators will be required to warm the air introduced for the ventilating requirements. Therefore:

$$H_v = 0.24\ W\ (t_y - t_o) \tag{6}$$

In this case t_y should be equal to or slightly higher than t. If the unit ventilator were of such capacity as to exactly provide for the ventilating requirements, the direct radiation would be selected on the usual basis. However, it is necessary to employ a unit which may not exactly meet the ventilating requirements, since standard units are usually rated in terms of the volume of air that will be delivered at a certain temperature t_y for an initial temperature of t_o. Therefore a certain amount of heat (H_h) may be available from the unit ventilator for heating purposes, as previously stated, and the amount of equivalent direct heating surface may, if desired, be deducted from the amount required for heating the room.

Example 2. Assume the same data as in Example 1, for a split system, and select the unit and direct heating surface, making allowance for any excess capacity of the unit above that required to warm the air introduced for ventilation purposes.

Solution. The amount of equivalent direct heating surface required, disregarding the unit ventilator, is equal to $\frac{24,000}{240} = 100$ sq ft. A unit having an air capacity of 1050 cfm is selected, the excess heating capacity of this unit for an entering air temperature of zero being 82 sq ft. The theoretical net direct heating surface required with the fan in operation is therefore 18 sq ft. However, it is not likely that this or any other unit contains enough indirect heating surface to heat the average size classroom when the fan is inoperative. The maximum deduction should not exceed 75 sq ft of equivalent direct heating surface, and as previously stated, the best results are obtained when no deduction is made for the excess capacity of the unit.

Chapter 14

TEMPERATURE AND HUMIDITY CONTROL

Definitions, Thermostats, Temperature Control Systems, Control of Automatic Fuel Devices, Zone Control, Control of Air Conditioning Systems, Control of Radiators and Convectors, Unit Heaters, Unit Ventilators, Central Fan Systems

CONTROL of a heating or cooling system may be obtained through regulation of only the dry-bulb temperature, while in an air conditioning system, the dry-bulb temperature, the wet-bulb temperature, and air movement must all be regulated. It is possible, however, that the control of only one of these may affect the other two sufficiently to give desired conditions.

This chapter contains information on the principles underlying the regulation of both temperature and humidity as well as data concerning various devices for such regulation. Specific control devices and systems are described in the Catalog Data Section of The Guide.

Controls are applied for the following reasons:

1. To maintain conditions required for human comfort and efficiency.
2. To maintain conditions required for industrial processes.
3. To obtain economy in operation.
4. To provide necessary safety measures.

The proper operation of all control systems depends on the selection of the correct type of control instrument, as well as on its correct application within the complete system.

DEFINITIONS

For the purposes of this chapter the terms used shall be construed as follows:

Normally Open or Normally Closed: Used to indicate position taken by a valve or damper when the power for the control system (compressed air, electricity, etc.) is turned off, or fails.

Branch Line: The line between a thermostat, humidistat or switch and the valve or damper it operates.

Main Line: A line furnishing power to a thermostat, humidistat or switch.

Pilot: A thermostat or humidistat which acts directly on another thermostat or humidistat. The pilot may either throw the instrument it operates out of action by cutting off or throttling its source of power, or it may act as a remote adjusting device by resetting its point of control. In the former case, the branch line of the pilot becomes the main for the other instrument. In the latter case, it is necessary for each to have direct connection to the main line.

Positive Acting: A thermostat, humidistat or relay which maintains its valve or damper either wide open or fully closed, that is, when there is always full force or none in its branch line.

Intermediate Acting: A thermostat, humidistat or relay which holds its valve or damper in any position, that is, when its branch line may have full main force, or any part of it.

Two-Point Thermostat: A thermostat having two branch lines, each of which is independent of the other, thus being equivalent to two separate instruments.

Direct-Acting Thermostat: A thermostat that increases power in its branch line on rising temperature. A *direct-acting valve* is one that is normally open.

Reverse-Acting Thermostat: A thermostat that increases the power in its branch line on falling temperature. A *reverse-acting valve* is one that is normally closed.

Threeway Valve: A valve having three pipe connections and used to regulate the flow of a fluid through either one of two circuits; it may be operated either positively or intermediately.

THERMOSTATS

Automatic control of temperature requires the use of an instrument known as a *thermostat* which responds to a change in temperature. Although there are many types of thermostats, the basic principles of operation of practically all types are included in the following classifications:

1. The *diaphragm* type (Fig. 1) which, by means of an expanding liquid or gas within a diaphragm or bellows, furnishes motion; which motion may be mechanically transmitted in proportion to the rise and fall of temperatures surrounding the diaphragm.

2. The *direct-expansion* type (Fig. 2) which operates by direct expansion and contraction of a substance which has a high coefficient of expansion such as hard rubber. The slight movement of the thermostatic element usually must be multiplied through a system of levers.

3. The *bi-metallic* type (Fig. 3 a, b, and c) which is actuated by means of two intimately attached metals having dissimilar coefficients of expansion. This type is inherently more sensitive to temperature changes than the diaphragm or direct expansion types, but lacks the necessary force to cause motion of the controlled equipment and must be used with compressed air, electric current or other source of power. The sensitive element may take any one of a number of forms, the most common being the straight strip, the circular strip, the spiral and the helix. Fig. 3-a shows the straight strip used in a compressed air thermostat, Fig. 3-b shows the spiral used in a mercury tube type of thermostat, and Fig. 3-c shows the curved strip used in an open contact thermostat.

Direct- and Indirect-Acting Thermostats

Thermostats may control directly, or indirectly, the equipment which regulates the supply of heat and are frequently classified on this basis as follows:

Direct-Acting or Self-Contained Thermostats. These instruments have sufficient power in themselves to actuate the controlling devices. They may be subdivided further into two groups: (1) those in which the sensitive element and the controlling device are in one body, the power from the thermostat being transmitted by a system of levers; and (2) those in which the sensitive element and the controlling device are separated, the power from the thermostat being transmitted through a capillary tube.

The expansion or volatilization of a fluid actuates a diaphragm which transmits its motion to a valve or damper. The small movement of the diaphragm may, if necessary, be multiplied and transmitted by mechanical means, such as a system of levers, or by liquid pressure in order to produce sufficient movement of the damper or valve. This thermostat is inherently intermediate acting since the expansion or volatilization of the fluid and the movement of the diaphragm are proportional to the change in temperature.

There are many types of thermostatic radiator valves which come in the first group.

In general they maintain some temperature at their location which bears a relation to the room temperature. Since these control devices are inherently intermediate acting, they should not be used on one-pipe systems because a partially closed valve will not allow the condensation to leave the radiator.

Fig. 4 shows the operation of a self-contained thermostat in connection with a hot water storage tank whereby the supply of steam to the tank is controlled directly by the temperature of the water in the tank. Fig. 5 shows the use of a self-contained thermostat on a hot water tank with vacuum return.

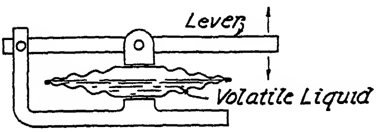


Fig. 1. Diaphragm Type Thermostat

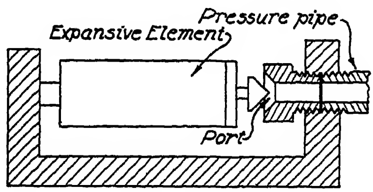
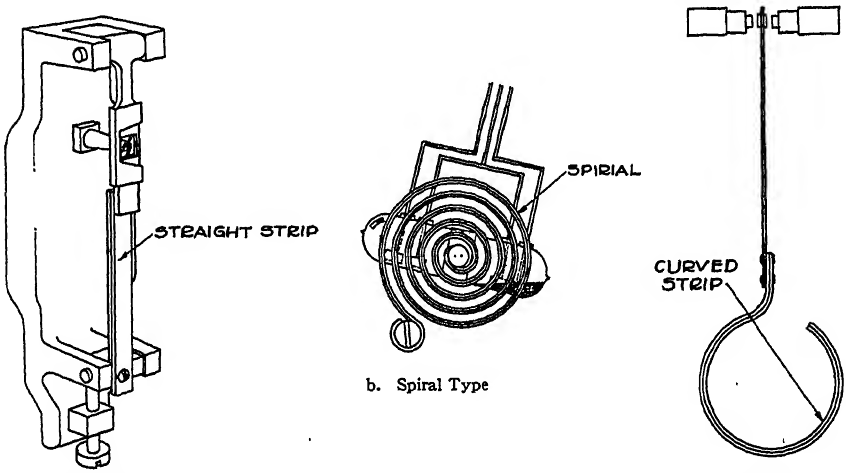


Fig. 2. Direct Expansion Type Thermostat



a. Straight Strip Type

b. Spiral Type

c. Curved Strip Type

Fig. 3. Three Types of Bi-Metallic Thermostat

Indirect-Acting Thermostats. These instruments do not have sufficient power to operate the controlling devices, but function to control the external source of power which in turn operates the valves and dampers. There are two distinct groups of these instruments, the *compressed air* and the *electric.*

Essentially the compressed air thermostat consists of a hole or leak port which is opened or closed by means of the thermostatic element. This port communicates with a diaphragm which in turn controls the supply of compressed air reaching the valve or damper operator. The instrument regulates the flow of the compressed air to and from the controlled devices. There are two classes of these instruments, one of which allows the air pressure to build up or be released instantly and is called the positive or snap-acting type; and the other which allows the compressed air to be maintained at any intermediate pressure, called the intermediate or gradual-acting type.

The electric thermostat makes or breaks one or more electric circuits. In other words it amounts to an automatically operated switch which opens and closes circuits in response to the change in temperature. The controlled devices may be constructed to be positive or gradual acting.

There are many models of all of these thermostats which may be generally grouped as follows:

1. *Room Thermostats.* All thermostats which are designed to be used in a room to control the temperature within that area.

2. *Duct Thermostats.* These instruments are constructed so that the thermostatic element is in the air duct or chamber and the remainder of the instrument is on the outside of the duct. This type of thermostat is generally used to control the temperature of the air in various parts of ventilating and air conditioning systems.

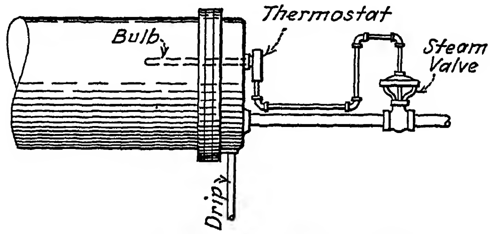


FIG. 4. SELF-CONTAINED THERMOSTAT IN HOT WATER STORAGE TANK

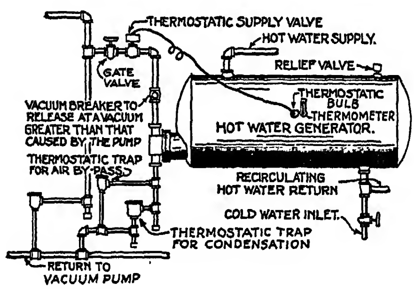


FIG. 5. SELF-CONTAINED THERMOSTAT ON HOT WATER TANK WITH VACUUM RETURN

3. *Two-Temperature Thermostats.* These instruments are used when it is desired to control at two distinct temperatures at different periods by means of a single thermostat. For example, a temperature of approximately 70 F may be desired during the day, while some lower temperature is satisfactory during the night. Shifting from the high to the low temperature may be accomplished at some remote point by means of a clock or a manual switch.

4. *Clock Thermostats.* These instruments are arranged with clock mechanisms to maintain predetermined temperatures at predetermined hours, such as a low night temperature which may be automatically increased at a certain hour to the proper day temperature.

5. *Weather Compensating Thermostats.* These instruments control the temperature of a building with respect to the outdoor temperature. Two thermostatic elements, one of which is placed outdoors and the other attached to a radiator, are arranged so as to operate the supply of heat in conjunction with each other. The temperature of the radiator is thus decreased as the weather temperature rises. This system may control the temperature of hot water, the pressure on a steam boiler, the volume of steam sup-

plied from outside the building, the amount of gas or oil fuel consumed, or the operating rate of a coal stoker.

TEMPERATURE CONTROL SYSTEMS

The foregoing discussion has concerned the sensitive element of the thermostat and its work. The following pertains to the types of control used for (1) direct radiators, (2) unit heaters, (3) unit ventilators, and (4) central fan systems.

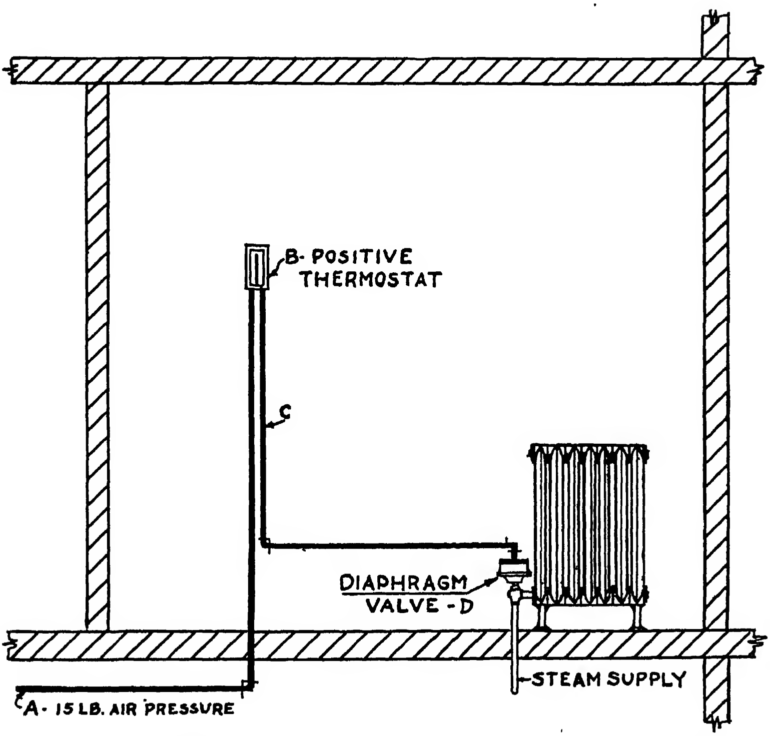


Fig. 6. Control of Direct Radiator with Positive Thermostat

Control of Radiators or Convectors

The control of rooms heated by radiators or convectors may be accomplished in several different ways. The problem is to control the heating units located at one or more points in or adjacent to the room. The circulation of air is not rapid, and the rate of temperature change usually is slow. The types of controls commonly used are: *first*, the direct-acting radiator valves with the thermostatic element at the valve, or at some adjacent point in the room and connected to the valve by means of a capillary tube; and *second*, the indirect-acting thermostat controlling either air or electrically operated valves. Fig. 6 shows a radiator controlled by an indirect-acting thermostat.

A discussion of control of steam heating systems is given in Chapter 31.

Control of Unit Heaters

Automatic control of unit heaters may be accomplished with and without by-pass units. There are two general methods of automatic control that may be applied to the unit without by-pass. One is to stop and start the unit heater motors individually or in a group by means of a room thermostat. The other is to control the steam supply either to individual units or to a group of units by means of a gradual-acting valve operated from a room thermostat. Where a steam valve control is used, the return system must be equipped with a vacuum pump, as an appreciable pressure difference between supply and return connections necessarily exists when the steam is throttled.

A combination of these two methods is sometimes used wherein the steam supply to the units is cut off completely before the unit heater motor is stopped and is turned on just before the motor is started again. The purpose of this control is to cool off the heater casing and to prevent possible discomfort to persons close to the unit from the radiant heat if steam is left on the coils when the motor is stopped. This is accomplished by two thermostats, one for controlling the motor and the other for controlling the steam valve. With this combination control, there is some possibility of draft complaint, for at times the fans may deliver cool air.

Gradual-acting control of the steam valve should not be applied to heaters with outside air connections unless some protection against freezing is also used. This protection can be accomplished by separating the heater coil into two parts, one of which acts as a tempering coil and is controlled from an outside air thermostat which will keep steam at full pressure on the coil whenever the entering air is below freezing, and which will cut off the steam when the entering air goes above freezing. The main coil is controlled from the room thermostat and is depended upon for temperature regulation. Another method of protection is to immerse in the return condensate a thermostat which controls a positive-acting by-pass valve and which will admit additional steam if the temperature of the condensate falls to the freezing point. This is purely a safety device and assumes that no condition of normal regulation will require enough throttling to make freezing possible. If this were not the case, such a method would not be suitable, as it prevents regulation beyond the point where the by-pass valve opens.

As a precaution against allowing the unit heater motors to continue to run if the steam supply fails or is for some reason shut off, either a pressurestat or thermostat in the supply line, or a thermostat on the return line may be installed to stop the motor when the pressure or temperature in the supply line, or the temperature in the return line, drops below a predetermined point.

The by-pass-type units are equipped with a by-pass opening and two interconnected dampers, so arranged that all or any portion of the air may be passed through or around the coil without being heated. It is possible to obtain a fair degree of temperature regulation by manually operating the dampers, but with the application of automatic controls, unusually good temperature control is obtained.

There are two types of control applicable to the by-pass units. The

older method is the *pneumatic* while the newer method is the *electric*. For the former, an air motor is installed on the unit to actuate the dampers. A pneumatic room thermostat is connected by means of small air lines to this air motor as well as to a source of compressed air supply. Should the room temperature tend to rise above the thermostat setting, the position of the dampers will shift to allow more air to be by-passed and less air to be heated through the coil. This produces a lower discharge temperature, which restores the desired temperature condition in the room. Conversely, should the room temperature tend to fall, the damper positions shift, so that less air is by-passed and more air is heated in going through the coil and a greater quantity of heat is supplied to the room.

The electric controls are similar to the pneumatic controls. In place of the air motor, a small reversible electric motor is used, and this is actuated by a room thermostat. When more heat is required, the thermostat makes a contact which causes the motor to shift the damper positions so that less air is by-passed and more air is drawn through the coil. When

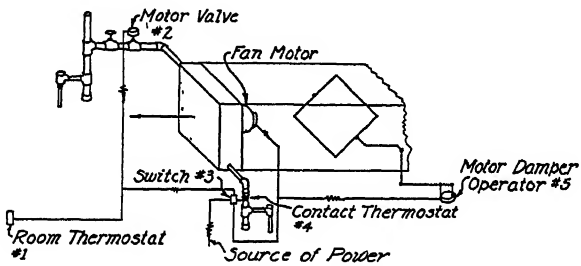


FIG. 7. ARRANGEMENT OF AUTOMATIC CONTROL FOR A UNIT HEATER

less heat is required, the thermostat makes another contact which causes the motor to operate in the reverse direction.

Where conditions require, an additional thermostat may be used to stop the fan motor when the by-pass damper is wide open and the coil damper tightly closed. Also, a thermostat may be applied to the return line which will stop the fan motor should the steam supply fail.

Fig. 7 shows a control for a unit heater which takes into consideration the factors which may arise. The room thermostat No. 1 calls for heat and passes electric current to motor valve No. 2, which operates, admitting steam to the heater. The same impulse from thermostat No. 1 also seeks to operate switch No. 3 and so to cause operation of the fan motor, but is prevented by contact thermostat No. 4 coiled around the return pipe of the heater. Thus unless the heater is warm its fan cannot operate and the ensuing cold drafts are avoided. As soon as the contact thermostat No. 4 becomes warm it permits operation of switch No. 3, starting the fan motor, and where desired also permits the motor damper operator No. 5 to open the intake damper from outside. The fan always stops and the damper always closes when the heater is cold.

Control of Unit Ventilators

The unit ventilator presents a different control problem than the unit heater. Generally this type of unit draws its supply of air from the outside, heats it, and introduces this air into the room under control. There are many types of unit ventilators on the market. Some have a mixing damper by which the temperature of the air entering the room may be varied, others have valves for this purpose, and still others use a combination of the two. Regardless of the construction of the machine, the essential requirement is that the temperature of the air delivered to the room should change slowly and remain as near room temperature as possible. Frequently direct radiators are used in conjunction with the unit ventilators to supply additional heat in extremely cold weather or for quickly heating up the room.

The four general types of control for unit ventilators are as follows:

1. A damper operator, the supply of power to which is controlled by a room thermostat, is attached to the mixing damper. When the thermostat calls for heat, the damper is moved to a position which forces more air through the heating unit and thus increases the amount of heat supplied to the room. This action must be gradual so that the air temperature may be changed slowly to prevent the drafty condition caused by supplying first hot and then cold air.

2. In mild weather the heating unit frequently supplies sufficient heat to cause overheating of the room, even though all of the air is by-passed around the heating unit. To avoid this fault a valve is placed on the heating unit to close the steam supply when the damper is by-passing all of the air. This valve is used in addition to the damper operator explained in the foregoing paragraph.

3. In some unit ventilators one or more heating units and no mixing damper are used. A gradual-acting valve on each heating unit controls the supply of steam to the unit to give the proper amount of heat required to maintain the desired room temperature. A thermostat to govern each valve may be installed in the room, or one thermostat may be used for all valves.

4. Another type of unit ventilator is arranged so that all recirculated air passes through the heating unit, and the outside air is introduced into the room for cooling purposes only. The outside air damper and the recirculated air damper are interlocked so that one damper operator will control them. In addition a valve operator is placed on the heating unit. Both of the operators should be gradual to avoid drafty conditions. When the thermostat calls for heat, the damper operator slowly closes the outside air damper and opens the recirculating damper simultaneously; if this does not meet the demand, the valve on the heating unit opens until the room temperature reaches the desired point.

Fig. 8 shows a unit ventilator which may or may not be used in conjunction with a direct radiator. It is generally considered advisable to shut off the steam to the direct heating surface, if any, first, so that the control system will allow the unit ventilator to supply the full heating requirement where possible. For additional information on the control of unit ventilators, refer to Chapter 13.

Central Fan Heating and Ventilating Systems

The numerous types of central fan systems present many control problems. In general they all have one point in common, namely, that the temperature change may be very fast due to rapid circulation.

System for Ventilating Only (Split System). Fig. 9 shows an accepted control for ventilating systems. Thermostat *A* located in the outside air

duct set just above freezing, controls a valve *C* on the first heating coil. This valve is either completely open or completely closed. The by-pass damper *B* and the other two valves *D* and *E* are controlled by a duct thermostat *F* located in the discharge duct from the fan. If the temperature of the air surrounding the thermostat *F* increases, the damper is moved to admit more cold air. Should this not reduce the temperature sufficiently, the valves on the heating coil will be closed gradually and in sequence until the correct temperature is reached. The opening or closing of the damper *B* and the valves *D* and *E* must be gradual or there will be a wide fluctuation in air temperature.

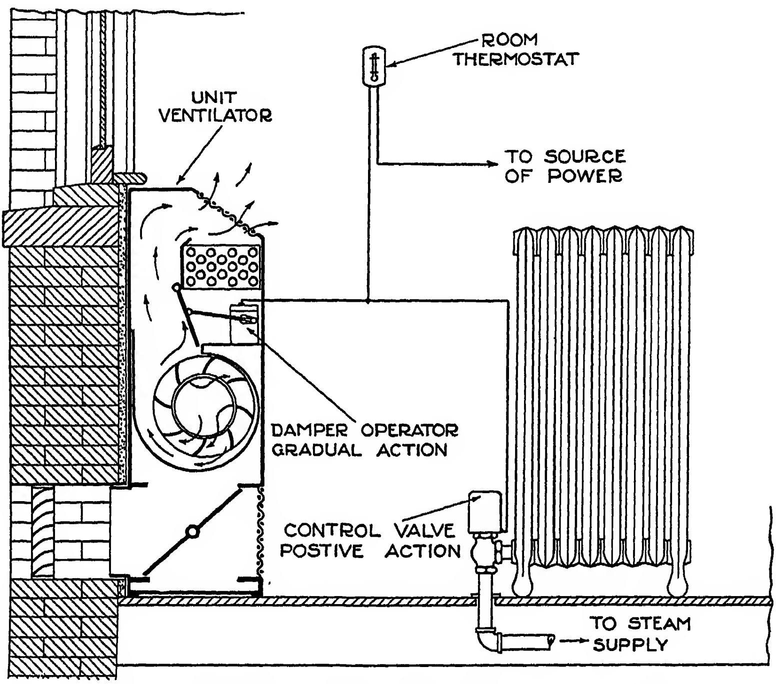


FIG. 8. CONTROL OF UNIT VENTILATOR AND RADIATOR

In ventilating systems it is customary to supply air to the ventilated spaces at an inlet temperature approximately equal to the temperature maintained in the rooms. The radiators therefore are designed to take care of all the heat losses from the room. Hence, in order to maintain controlled room temperatures it is further necessary to use room thermostats governing control valves placed on the radiators. With this type of central fan system it is possible to ventilate a large number of rooms by means of one fan.

In some installations, such as in theaters or auditoriums, it is difficult to install sufficient direct heating surface to offset the heat losses from the room. Also there are installations where a short heat-up period is allowed before occupancy of the room, and it is advisable to use the

entire heating capacity of the ventilating system for this purpose. Systems of this type have, in addition to the control described in Fig. 9, a pilot thermostat located in the room to be heated or in the path of air with-

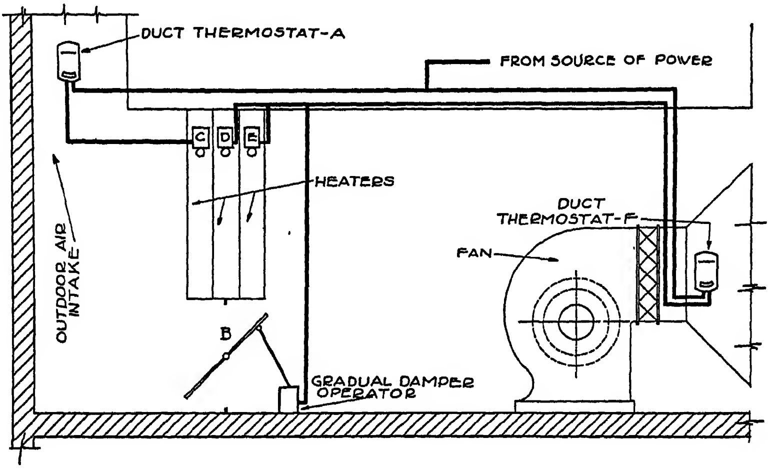


FIG. 9. CONTROL OF A SPLIT SYSTEM OF VENTILATION

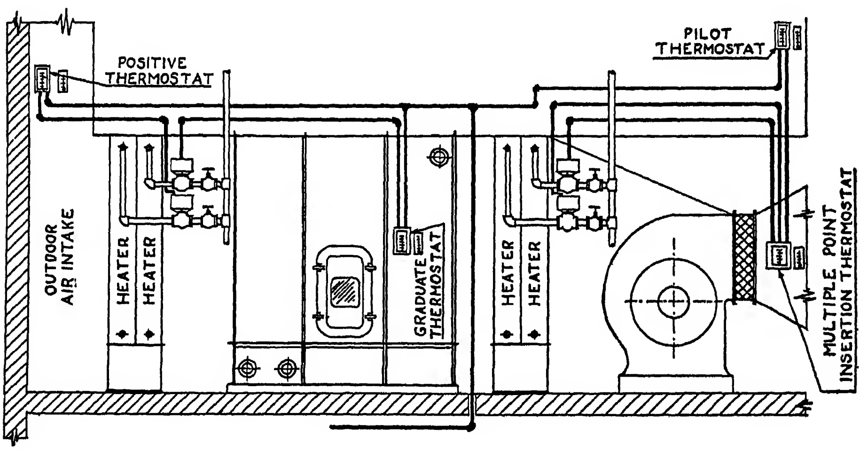


FIG. 10. USE OF PILOT THERMOSTAT ON VENTILATING SYSTEM WITH AIR WASHER

drawn from that room. This thermostat is set for the temperature required in the ventilated space and controls the supply of power to the thermostat *F* located in the fan discharge controlling the damper *B* and the heating units *D* and *E*.

When the temperature of the air in the room is below the setting of the pilot thermostat, the control is such that the by-pass damper *B* is closed and the valves on the heating coils are open. Thus the maximum heating capacity is available to bring the room temperature to the desired point.

When the room temperature reaches the setting of the pilot thermostat, the control of the air temperature is returned to the thermostat F in the fan discharge. The fan discharge thermostat, therefore, serves as a minimum temperature control and prevents introduction of air at temperatures which would produce drafts or faulty air distribution. Accurate control of room temperature is accomplished in this manner, but the system can not well be used where more than one room is supplied with air from a single fan.

In central fan systems, air washers are often used and in such cases, due to the effect of temperatures on humidity, additional control is required. Fig. 10 shows such an arrangement with control of the second tempering

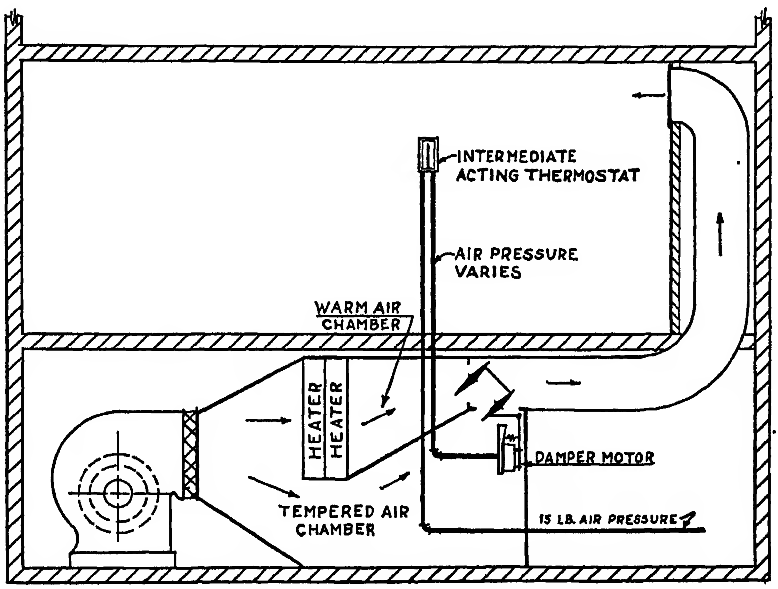


FIG. 11. CONTROL OF MIXING DAMPERS WITH INTERMEDIATE ACTING THERMOSTAT

heating unit from the air washer temperature and with the usual control of the first tempering heating unit from the outside temperature. This permits the air to be kept cool while passing through the washer so that too much moisture will not be absorbed. Fig. 10 also shows control of the re-heating units from a duct thermostat in the fan discharge and the application of a pilot thermostat to a system of this sort.

Combined Systems. There are various central fan systems which are used for both heating and ventilating. They are usually arranged with tempering heating units, automatically controlled to provide a minimum temperature for ventilating only, and additional heating units to supply the heating requirements. Fig. 11 shows a type of system which has the reheating units located in the fan room. Tempered air at about 70 F is supplied to the fan and may be further heated by the reheating units, or it may pass into the tempered air chamber. A room thermostat controls a

gradual-acting damper operator on the double mixing damper in the warm and tempered air chambers. When the thermostat calls for heat, the damper operator moves the dampers so that more air is taken from the warm air chamber. It is essential that the double mixing damper be moved slowly to prevent alternate blasts of hot and cold air from being supplied to the room.

Outside Air, Recirculating and Vent Dampers. In all types of plenum systems, the outside air damper is usually opened and closed by a damper operator. This operator may be controlled from a switch in the engineer's room or it may be operated by a relay in the fan motor circuit. When the ventilating fan is started, the relay causes the damper operator to open the outside air damper.

Recirculating dampers and vent dampers may also be opened and closed by means of damper operators controlled from remote locations. Generally these damper operators are positive acting and are either completely opened or closed. However, in some cases where part outside air and part recirculated air is used, it is advantageous to use damper operators which have a certain number of definite positions. With this type of operator it would be possible to use 75 per cent outside air and 25 per cent recirculated air, or any other proportions which might be predetermined. These damper operators are controlled from switches generally mechanically interlocked so that the total opening of the two dampers is 100 per cent.

Hand-Fired Coal Systems

In small buildings the heating plant may be controlled by a single thermostat located in a key room in the building, instead of each room having its own control.

The most common control for a hand fired furnace or boiler consists of a room thermostat and a furnace regulator of some type. The thermostat should be located in a representative room; never, of course, near the chimney or heat flue, too close to a radiator, or in a drafty hallway, and preferably on an inside wall. The regulator is attached to the draft and check dampers of the furnace. When the temperature of the air surrounding the thermostat drops, the thermostat causes the furnace regulator to open the draft and close the check damper. As soon as the room comes up to temperature, the draft is closed and the check damper opened. With this arrangement on hot water heating systems it is advisable to install an immersion thermostat in the boiler. This thermostat should be connected with the room thermostat so that both must call for heat before the draft is opened, but either one may cause the draft to be closed. On warm air systems it is advisable to use a bonnet thermostat and on steam heating systems a pressure limiting device, in series, in each case, with the room thermostat. If the temperature of the heating medium becomes too high, the drafts will be closed even though the room thermostat continues to call for heat.

CONTROL OF AUTOMATIC FUEL APPLIANCES

It is essential that automatic temperature control be used with oil burners, gas burners, and stokers to aid economical operation. There are

many types of burners and many types of control, but there are some points common to all. First, a room thermostat is located in a key position in the building to maintain a given temperature at that point.

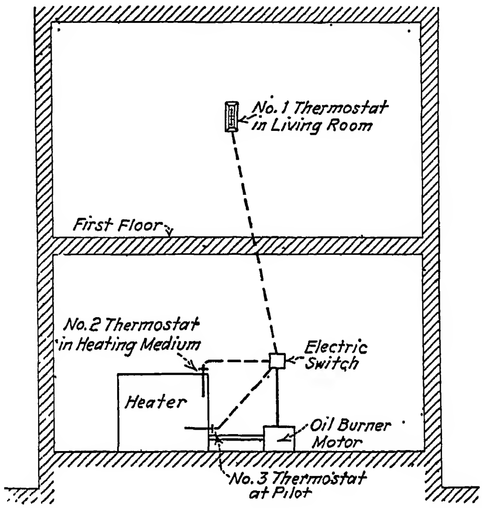


FIG. 12. ELECTRIC THERMOSTAT APPLIED TO OIL FIRED HEATING SYSTEM

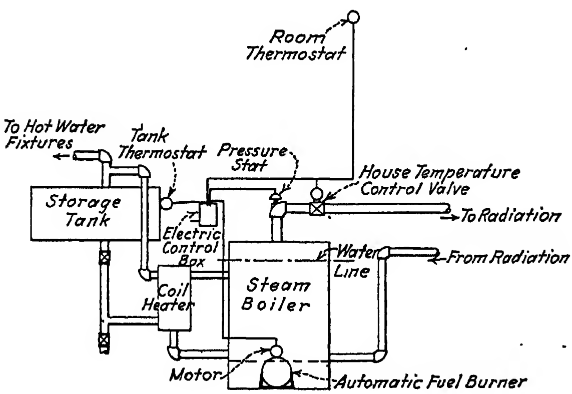


FIG. 13. TYPICAL ARRANGEMENT OF STEAM OR VAPOR SYSTEM WITH TWO THERMOSTATS CONTROLLING AUTOMATIC FUEL BURNER USED FOR HOUSE HEATING AND WATER HEATING

Safety devices are installed in connection with this thermostat so that a failure of the ignition, power, or fuel supply will shut the system down. The same limit controls as recommended for coal burning should be used.

Oil Burners

Fig. 12 illustrates diagrammatically the essentials of an oil burner control circuit. Three thermostats are employed as shown in the illustration. Thermostat No. 1 will stop the burner when the room temperature is too high and No. 2 will stop the burner when the temperature of the heating medium exceeds the setting of thermostat No. 2. Both temperatures must be below their respective thermostat settings to start the burner. Thermostat No. 3 responds to the flame temperatures and in conjunction with the control switch acts as a safety to stop the burner if the latter fails to ignite or burn properly as demanded by thermostats Nos. 1 and 2.

Steam and hot water heating plants are often used to provide heat for the domestic hot water supply as well as for heating the building. Fig. 13 illustrates one such system. The burner control is similar to that shown in Fig. 12 except that either the room thermostat or the tank thermostat may cause the opening of the valves in the mains which supply them and start the fuel burner, but the burner will not stop unless both thermostats have closed their valves or the steam pressure shall have reached that allowed by the pressurestat. Much the same control is applied to gas burners and automatic coal stokers.

Gas Heating Appliances

On account of the ease and effectiveness with which the fuel can be controlled, gas-burning appliances are particularly adaptable to full automatic control. Standard equipment on a steam boiler generally includes provision for control through a room temperature thermostat, a steam pressure regulator and a device which shuts off the gas in the event that the water level becomes too low. Practically all gas boilers are or may be equipped with automatic safety pilots which shut off the gas if the pilot flame is too low.

Water boilers are adapted to operation under thermostatic room temperature control and are also provided with water temperature control equipment. Warm air furnaces can be under the control of thermostats in the spaces being heated, as well as thermostats located in the heat ducts for the purpose of preventing unpleasantly hot air reaching the heated spaces. Variations in the pressure under which the gas is supplied to the appliance are controlled by means of a gas-pressure regulator. This is an essential part of practically all makes of gas-burning heating appliances, in fact, a gas-pressure regulator is required by the *American Gas Association* on all approved gas boilers, warm air furnaces (except floor furnaces) and unit heaters.

ZONE CONTROL

Zone control is a step between a single thermostat and individual room temperature control. The building is first divided into sections or zones which may have quite different heat requirements. With this method of control:

First: The zoning should be done with reference to the compass, since the north and west quarters in most localities require considerably more heat during the heating season than do the south and east quarters.

Second: Most large office buildings have more or less space occupied by merchants, and some by clubs, restaurants, etc., which have short hours of occupancy. Much can be accomplished in zoning with reference to the kind of occupancy of space. For additional information on this subject, refer to Chapter 31.

COOLING UNITS

Cooling units are readily adaptable to thermostatic control. Several arrangements are as follows:

1. Room thermostat in conjunction with a magnetic or motor-operated valve to regulate the flow of refrigerant to coil. Usually the fans operate continuously.
2. Room thermostat to control operation of compressor. Fans operate continuously.
3. Room thermostat to control the operation of the fan motors.
4. Room thermostat to control the operation of fan motor and compressor motor simultaneously.
5. Room thermostat to control operation of the compressor with back pressure control to regulate the fans.

For further information on unit coolers, see Chapter 12.

INDUSTRIAL PROCESSES

There are many industrial processes requiring automatic temperature and humidity regulation. The control equipment operates on the same principles that have been described, but is often specially designed for each particular process. Each installation, or the installation for each process, is likely to be a problem peculiar to that process.

AIR CONDITIONING SYSTEMS

The following fundamental principles should be borne in mind in the solution of problems involving the control of air conditioning systems:

1. Dew-point temperatures vary only with the amount of moisture. That is, no matter how much a given mixture of air and water vapor is heated or cooled, the dew-point temperature remains the same, as long as there is no addition or subtraction of water. Cooling below the dew-point temperature will, of course, cause subtraction. *Also,* at the same temperature, there is always the same proportion of water vapor in the saturated mixture, provided sufficient water and time are furnished for saturation.

Table 2, Chapter 1, shows the amount of moisture required to saturate a space at various temperatures. When the proper amount of moisture is determined, it is only necessary to set the air washer (dew-point) thermostat for the corresponding temperature of saturation; then if the air entering the washer has more humidity than desired, the excess will be condensed; and if it has less, the deficiency will be absorbed from the sprays.

For example, the dew-point temperature at 70 F and 40 per cent relative humidity is 45 F. Therefore, if the air temperature is maintained at 45 F as it leaves an air washer (assuming it is fully saturated) and then is heated to 70 F, it will have a relative humidity of 40 per cent. If it is desired to maintain these conditions in a given space, the air temperature can be raised to any necessary point, say 120 F (at which the relative humidity will be only 9 per cent). When the heat in the air has been dissipated through the walls, roof, etc., the space temperature being maintained at 70 F, the relative humidity will be 40 per cent.

2. Within ordinary operating ranges, saturated air will have a relative humidity of approximately 50 per cent when its temperature is raised 20 deg. For example, satu-

rated air at 40 F raised to 60 F has a relative humdity of 49 per cent; 60 F saturated air raised to 80 F has a relative humidity of 50 per cent. (See Table 1, Chapter 1). Thus a differential thermostat can be used to maintain a nearly constant relative humidity of 50 per cent by holding the dew-point temperature 20 deg below the dry-bulb temperature.

3. The *total heat* of the air and water vapor mixed with it varies directly with the wet-bulb temperature. For example, the occupants of an auditorium give off sensible heat which tends to raise both the dry-bulb and wet-bulb temperatures of the space; but they also give off moisture which increases the absolute humidity and tends to further raise the wet-bulb temperature an amount which is a direct indication of the heat expended by the body in evaporating this water. This relationship is useful in regulating the total heat, as wet-bulb temperatures can be controlled directly by means of a thermostat having a sensitive element covered with water-fed wicking, similar to a wet-bulb thermometer.

For example, the total heat of air at 80 F and 60 per cent relative humidity is the same as for air saturated at 70 F, *i.e.*, 33.5 Btu per pound, both having a wet-bulb temperature of 70 F. Air at 80 F and 60 per cent relative humidity (70 F wet-bulb = 33.5 Btu per pound) reduced to 70 F and 50 per cent relative humidity (= 58½ F wet-bulb = 25.2 Btu per pound, total heat) must give up 8.3 Btu per pound. If the sensible heat and moisture pick-up in an auditorium is 8.3 Btu per pound of air handled in the conditioning system, the wet-bulb temperature of the air entering the space must be maintained at 58½ F to secure a final condition of 80 F and 60 per cent relative humidity.

Control of Relative Humidity

Relative humidity is controlled by means of instruments called humidistats or hygrostats, or by the proper combination of two or more thermostats. The following are the most commonly used methods:

1. A thermostat is located in or at the outlet of a spray-type air conditioner which maintains a constant saturation temperature of the air leaving the conditioner by varying the temperature of water entering the suction of the pump supplying the spray nozzles, or by varying the temperature of the air entering the conditioner, or both. The temperature of the air entering the conditioner may be varied by use of tempering heaters, or by the proper proportioning of supply and return air entering the conditioner. This thermostat is known as a dew-point thermostat, as it determines the dew-point temperature of the air introduced into the conditioned spaces. A second thermostat in the room, or in the path of the air leaving the room, maintains a constant dry-bulb temperature by varying the amount of sensible heat added to the air leaving the conditioner, or by varying the volume of air introduced into the conditioned spaces. These two thermostats, in combination, control the dry-bulb and dew-point temperatures, which accordingly fix the relative humidity.

2. A wet-bulb thermostat is located in the room, or in the path of the air leaving the room, to maintain a constant wet-bulb temperature by varying the saturation temperature at the air conditioner outlet. A dry-bulb thermostat is located in the room to maintain a constant dry-bulb temperature, which in combination with a constant wet-bulb temperature fixes the relative humidity.

3. A differential thermostat may be used to control relative humidity. This instrument consists of two thermostatic elements, one of which is in the path of the air leaving the conditioner, and the other under the influence of the dry-bulb temperature in the room. Instruments of this kind maintain a constant relative humidity by maintaining a constant difference between the dew-point temperature and dry-bulb temperature in the room. (See Item 2 under Air Conditioning Systems). One thermostatic element may be equipped with a moistening device to permit it to operate on wet-bulb temperatures. Such an instrument can be used to control the wet-bulb depression and thus the relative humidity.

4. A humidistat which responds directly to changes in humidity may be used to maintain a predetermined relative humidity with constant or with varying temperature. It may do this: by varying the dew-point temperature of air leaving a conditioner; by varying, with dampers, the proportion of moist and dry air; by varying the amount of moisture otherwise added to the air; or by varying the dry-bulb temperature.

Humidification for Residences

The principles underlying humidity requirements and limitations for residences are summarized in *University of Illinois Bulletin No. 48*[1], as follows:

1. Optimum comfort is the most tangible criterion for determining the air conditions within a residence.

2. An effective temperature of 65 deg[2] represents the optimum comfort for the majority of people. Under the conditions in the average residence a dry-bulb temperature of 69.5 F with relative humidity of 40 per cent is the most practical for the attainment of 65-deg effective temperature.

3. Evaporation requirements to maintain a relative humidity of 40 per cent in zero weather depend on the amount of air inleakage to the average residence, and vary from practically nothing to 24 gal of water per 24 hours.

4. Relative humidity of 40 per cent indoors cannot be maintained in rigorous climates without excessive condensation on the windows unless tight fitting storm sash or the equivalent are installed.

5. The problems of humidity requirements and limitations cannot be separated from considerations of good building construction, and the latter should receive serious attention in the installation of humidifying apparatus.

The following conclusions were drawn from the experimental results reported in the aforementioned bulletin:

1. None of the types of warm air furnace water pans tested proved adequate to evaporate sufficient water to maintain 40 per cent relative humidity in the Research Residence except only in moderately cold weather.

2. The water pans used in radiator shields tested did not prove adequate to maintain 40 per cent relative humidity in a residence similar to the Research Residence when the outdoor temperature approximated zero degrees Fahrenheit.

Central Fan Air Conditioning Systems

In central fan air conditioning systems as described in Chapters 9 and 22, varying amounts of outside and recirculated air are used, except where contamination prevents re-use, and in general, heat is supplied after the air washer, for obtaining humidity control under winter conditions. There are many control variations in use, and Fig. 14 shows a composite diagram, rather than a system of control for a single installation. The control valves for a dehumidifying air washer are shown in Fig. 15. The functions of the control devices shown in Figs. 14 and 15 are as follows:

Winter Operation (with steam)

1. Direct-acting thermostat *A* opens direct-acting valve in steam supply to a low-capacity tempering coil *P;* in sub-freezing weather it is set at 35 F.

2. Direct-acting thermostat *B* in the path of air leaving the second tempering coil *Q* controls direct-acting valve in steam supply to the coil *Q* at 45 F.

3. Direct-acting thermostat *C* controls normally-closed intake *M* and normally-open return air *N* dampers at 50 F. This location of thermostat *C* is primarily for operation with steam heating and at such times as by-pass damper *O* is closed. See discussion under heading *Spring and Fall Operation.*

[1]See Humidification for Residences, by A. P. Kratz (*University of Illinois, Bulletin No.* 48).

[2]66 deg is the optimum winter effective temperature recommended by the A.S.H.V.E. Committee on Ventilation Standards. See Chapter 2.

4. Humidistat or wet-bulb thermostat *D* in return air, acting through relay, causes *C* to take outside air when the relative humidity rises above 55 per cent (or the wet-bulb temperature rises above 60 F); also, if necessary, shuts off the water supply to the spray heads in the air washer and opens the supply to the flooding nozzles at the eliminator plates, by operating threeway valve *U* (Fig. 15). The relative humidity must, of course, be changed to suit the requirements. It must be maintained low enough to avoid condensation on walls or windows[3].

5. Reverse-acting thermostat *E* in discharge end of air washer operates a threeway valve (*V*, Fig. 15) in water circulating line, so as to cause water to pass through or around a heating unit in order to produce the correct dew-point temperature by adding any necessary heat to the water. It may also operate reverse valve *W* (Fig. 15) in the steam supply to the heating unit. The heat added may be only that sufficient to make up the temperature drop through the washer, due to evaporation. This thermostat is reverse-acting to prevent over-humidification in case of failure of motive power.

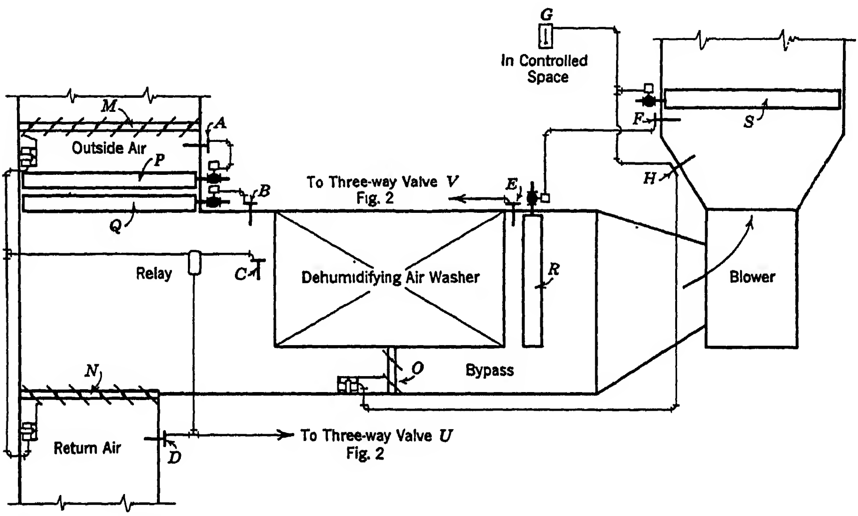


FIG. 14. DIAGRAMMATIC ARRANGEMENT OF VARIOUS PHASES OF CONTROL FOR A CENTRAL FAN AIR CONDITIONING SYSTEM

6. Direct-acting thermostat *F* in fan discharge operates a direct-acting valve in steam supply to heater *R* to produce the lowest temperature at which air can be introduced into the conditioned space, without complaints of draft. This varies from 60 to 70 F, depending on the velocity through, and location of, the supply grilles.

7. Direct-acting room thermostat *G* in a representative location controls direct-acting valve in steam supply to coil or coils *S* which supply the heat to replace the loss from the conditioned space.

Summer Operation (with refrigeration)

Thermostats *A*, *B*, *F* and *G* all hold their valves closed during summer temperatures which are above the thermostat settings, although this is unimportant while no steam is being supplied.

1. Thermostat *C*, having been set for 50 F, supplies power to open wide the intake damper and close the return air damper under the higher summer temperatures and this

[3]See discussion of condensation in Chapter 7. Also see paper entitled, Frost and Condensation on Windows, by L. W. Leonhard and J. A. Grant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

power can be passed through a graduating switch to permit manual operation of the dampers. As the wet-bulb temperature (or total heat) of the outdoor air is now normally greater than that of the return air, it is desirable, in order to keep down cooling costs, to recirculate the maximum amount of air.

2. Humidistat *D* is by-passed so that main power is applied direct to threeway valve *U* (Fig. 15) to prevent shutting off sprays. This by-pass can be arranged for cutting in manually, or automatically, with the starting of the refrigerating machinery.

3. Thermostat *E*, operating threeway valve *V* (Fig. 15), now determines whether the spray water is passed through refrigerated coils or is recirculated without treatment, and thus regulates the dew-point temperature. It is assumed that steam and refrigeration are not both turned on at the same time.

4. Direct-acting thermostat *H* operates a normally open damper *O* in the by-pass space around the air washer so as to mix the warmer return air with the cold air leaving the dehumidifier in such proportions as to give the minimum temperature at which air can be introduced into the conditioned space. This might be 70 F for a room temperature of 85 F. A switch should be installed in the branch line from *H*, and so connected to a main line as to permit keeping normally-open damper *O* closed during winter operation.

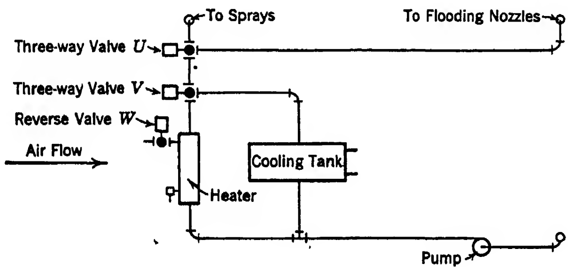


FIG. 15. CONTROL VALVES FOR A DEHUMIDIFYING AIR WASHER

5. Thermostat *G*, in addition to operating valve on heating unit *S*, acts as a pilot for thermostat *H* so as to retard the action of the latter in closing the by-pass damper, when the space temperature is below the desired point.

Spring and Fall Operation

During a considerable part of the year, conditioning can be accomplished merely by using all outside air or by mixing it with returned air. For example, when the total sensible heat gain in an auditorium is 2.4 Btu per pound of air being treated, outside air will be raised from 60 F to 70 F by the heat gain. During this period when dry-bulb temperatures are to be maintained at, or not much above, 70 F, the gain in sensible heat is the only factor that need be considered, because it is large in comparison with the gain in latent heat, except in restaurants and some classes of industrial work. The intake and recirculating dampers can then be operated by thermostat *F* set at 60 F. (It is assumed that such an outlet temperature can be used; if not, the volume of air should be increased). Thermostat *H*, being set higher for hot weather, holds by-pass damper *O* open to provide a maximum volume of air. In order to minimize overhumidification, the air washer and by-pass are arranged so that the return air stream tends to use the by-pass. However, since dehumidification is not required, as previously stated, the humidity control is obtained by shutting off the spray water by humidistat *D*.

Except for heating-up periods or other times when the heat gain is not greater than the heat loss, a system of this type can be operated without artificial heat, with outdoor temperatures as low as 40 F. For this reason it is economical to place a thermostat in the return air near *D* set to shut off a diaphragm valve in the main steam supply to the system at a temperature about 3 deg below that desired in the conditioned space. A pilot thermostat exposed to the outdoor temperature prevents the shut-off on days colder than 40 F.

As previously stated, there can be many variations from these descriptions, some of which are:

1. Tempering coils may consist of only one bank, *P* or *Q*, controlled by either *A* or *B* thermostat. In any case the capacity of the heating unit controlled by the outdoor temperature must be as low as feasible, otherwise if steam is supplied to it when the outdoor temperature is 30 F, the temperature of air entering the washer is likely to be too high to permit maintaining the proper dew-point temperature.

2. Both tempering coils may be omitted and return air mixed with outside air by

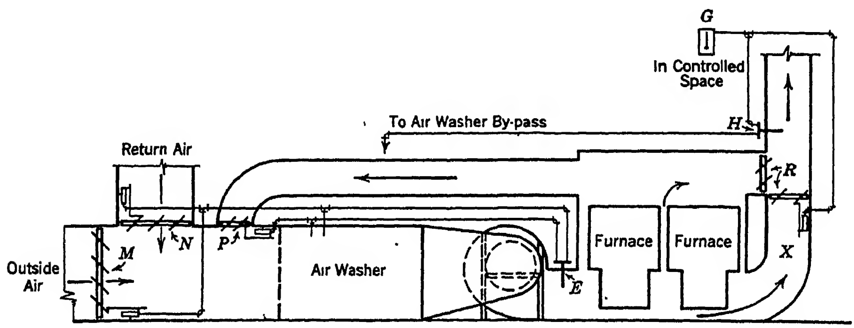


FIG. 16. DIAGRAMMATIC ARRANGEMENT OF CONTROL FOR A FAN-FURNACE AIR CONDITIONING SYSTEM

thermostat *C* so as to provide a proper temperature at the washer inlet. In this case, humidistat *D* should not act as a pilot.

3. The heating unit for the air washer water may be omitted, and the proper dew-point temperature maintained by placing thermostat *C* in the location of *E*. This requires additional heat from the tempering coils, or more return air to make up the loss due to evaporation in the washer.

4. Heating unit *S* may be combined with *R* in one or two banks and controlled by a one- or two-point thermostat at *F*, set for the minimum temperature at which air can be admitted into the conditioned space. For heating purposes, thermostat *G* then becomes a pilot for *F* so that these heating units are operating at full capacity when the space is cold, and are throttled by *F* when no heat is required.

5. A better arrangement than that described in the preceding paragraph is the use of an automatically readjustable thermostat at *F*, which can operate at any temperature between a proper minimum and a necessary maximum, depending on the temperature of the space. Thus for winter operation when the room temperature is 68 F, the blower delivers air sufficiently warm to supply the heat required under extreme conditions, and when it is 74 F, the delivery will be as cool as possible without complaint of drafts. A similar instrument can be used to replace *H*, and set to operate between 60 and 80 F for summer conditions.

6. For summer use, a remote readjustable thermostat can be located at *H*, and can be reset by a pilot exposed to the outdoor temperature. Thus as the outdoor temperature increases, the space temperature is maintained at a higher point.

7. A constant portion of the return air may be brought to a point between the air washer and the blower, and the temperature of the air leaving the washer regulated to give the proper result at *H*. The regulation is accomplished by shutting off one or more groups of sprays, or by changing the temperature of the spray water until the proper degree of cooling is secured.

8. Where an air washer is selected large enough to pass all the air handled by the fan, the by-pass and its damper *O* are not used. The washer sprays must then be divided into two side-by-side sections so that one section can be turned on or off by *H* to provide the proper temperature.

9. Where an ejector type heating unit is used for the spray water, a reverse-acting valve similar to *W* (Fig. 15) must be placed in the steam supply to be operated by thermostat *E*. In this case it is usual to install in this steam line another reverse-acting diaphragm valve to be operated directly by the water pressure in the pump discharge line. This automatically shuts off the steam when the water circulating pump is not in operation.

10. Based on the fact that the spray water in the air washer pan has practically the same temperature as the air leaving the washer, dew-point control can be accomplished by installing thermostat *E* in the water pan.

11. Where cold well-water is used for dehumidification, it is admitted to the sprays through a threeway valve similar to *V*, operated by thermostat *E*.

12. Control of steam heat is shown entirely by valves, although it is usual to install a by-pass damper around each heating unit and operate it, either with or without a damper over the face of the heating unit, in conjunction with the valve.

Capacities to be Selected

In designing apparatus to be automatically controlled, it is advisable to select minimum capacities for desired results. This not only assists the control system, but tends to produce desirable economies in initial costs. For example, in Fig. 14 the heating unit *R* selected should be one having a capacity that will just raise the temperature from the lowest desired dew-point (about 40 F) to the highest dry-bulb temperature required at *F* (about 70 F). Similarly, heating unit *S* should be of sufficient capacity to raise the temperature only enough to provide the heat necessary for the space to be conditioned.

Dampers should be sized so that extreme conditions will require the full open and closed position. For example, if it is desired that some minimum amount of outside air be supplied constantly during operation of the plant, damper *M* should be divided into two parts, one of which is operated by the thermostat and the other, large enough to supply the constant minimum, is operated by a separate positive-acting switch.

FAN-FURNACE AIR CONDITIONING SYSTEMS

The theory of control of mechanical warm air or fan furnace systems described in Chapter 23 is the same as for other types of central fan systems, but the methods are somewhat different. Fig. 16 shows the arrangement for winter operation.

Winter Operation

1. Thermostat *E* is a two-point instrument with one direct- and one reverse-acting relay. The former operates the normally-closed damper *M* and normally-open damper *N*, thus mixing return air to obtain the proper temperature (40 to 50 F) at the discharge

end of the washer. When this source of heat is insufficient, the reverse-acting relay opens normally-closed damper *P* (Fig. 16) to admit air from the warm-air chamber.

2. Direct-acting thermostat *G* operates double damper *R* (Fig. 16) to mix heated air with that coming direct from the washer through passage *X*, to obtain the proper temperature for delivery into the conditioned space. The dew-point temperature produced at *E* is not affected by passing some of the air through the furnaces, and the proper relative humidity is thus secured.

Summer Operation

3. Thermostat *E*, through its reverse-acting relay, operates a threeway valve (such as *V*, Fig. 15) to regulate the cooling effect on the spray water; or, through its direct-acting relay, it operates another threeway valve (such as *U*, Fig. 15) to shut off the sprays when the temperature is too low.

4. Thermostat *H* moves a dehumidifier bypass damper (not shown) in conjunction with *G* as a pilot, as described for a steam system.

Additional information on the control of fan-furnace systems will be found in Chapter 23.

Chapter 15

AIR POLLUTION

Sources of Air Pollution, Effects of Air Pollution on Health, Pulmonary Effects, Occlusion of Solar Radiation, Industrial Air Pollution, Abatement of Atmospheric Pollution, Smoke Abatement, Dust and Cinder Abatement

THIS chapter considers the hygienic aspects of atmospheric pollution and the methods by which this pollution may be lessened. Information concerning the cleaning of air brought into buildings for ventilating purposes will be found in Chapter 16, and a discussion of the exhausting of dusts and toxic gases from factories and industrial plants is considered in Chapter 21.

SOURCES OF AIR POLLUTION

The impurities which contribute to atmospheric pollution include carbon from the combustion of fuels, particles of earth, sand, ash, rubber tires, leather, animal excretion, stone, wood, rust, paper, threads of cotton, wool, and silk, bits of animal and vegetable matter, and pollen. Microscopic examination of the impurities in city air shows that a large percentage of the particles are carbon. (See Fig. 1, Chapter 16, for size of impurities in air).

Dust, Fumes, Smoke

The most conspicuous sources of atmospheric pollution may be arbitrarily classified according to the size of the particles as dusts, fumes, and smoke. *Dusts* are particles of solid matter varying from 1.0 to 150 microns in size. *Fumes* include particles resulting from chemical processing combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size. *Smoke* is composed of fine soot or carbon particles, less than 0.1 micron in size, which result from incomplete combustion of carbonaceous materials, such as coal, oil, tar, and tobacco. In addition to carbon and soot, smoke contains unconsumed hydrocarbon gases, sulphur dioxide, sulphuric acid, carbon monoxide, and other industrial gases capable of injuring property, vegetation, and health.

The lines of demarcation in these three classifications are neither sharp nor positive, but the distinction is descriptive of the nature and origin of the particles, and their physical action. Dusts settle without appreciable agglomeration, fumes tend to aggregate, smoke to diffuse. Particles larger than one micron will eventually settle out by gravitation; particles smaller will remain in suspension as permanent impurities unless they agglomerate to sizes larger than one micron.

Fly-Ash, Cinders

The term *fly-ash* is usually applied to the extremely small particles of ash, and the term *cinder* to the larger particles of coke and ash which are discharged with the gases of combustion from burning coal.

EFFECTS OF AIR POLLUTION ON HEALTH

Many kinds of dusts and gases are capable of producing pathological changes which may cause ill health. The harmful effects depend largely upon the chemical and physical nature of the impurities, and the concentration, length of time, and conditions under which they are breathed. Dust particles must be minute in size to be inhaled at all, although fairly large particles may gain access to the upper air passages.

The human body possesses remarkable filtering media for protecting the lungs. Small hairs which line the nasal passages, and a multitude of microscopic hairs, called *cilia*, in the epithelium lining in the bronchial tubes intercept many of the dust particles before they reach the lungs.

PULMONARY EFFECTS

The constant inhalation of dusts in city air irritates the mucous membranes of the nose, throat, and lungs, and eventually may produce discomfort and a series of minor respiratory disorders. The pigmented lung of the city dweller is an example of the pathological change produced over a period of years. This condition may be of no clinical importance, but an exaggeration of it in the coal miner results in anthracosis or dark spots on the lung due to the presence of phagocyted pigment in the lymph channels which impairs the functioning of the lung cells under stress.

Effects of Solids

Bronchitis is the chief condition associated with exposure to thick dust, and follows upon inhalation of practically any kind of insoluble and noncolloidal dust. Atmospheric dust in itself cannot be blamed for causing tuberculosis, but it appears to have a marked influence in aggravating the disease once it has started. There is, however, quite reliable evidence that carbon pigment, one of the atmospheric dusts, tends to wall off local tuberculosis rather than to further its spread.

The sulphurous fumes and tarry matter in smoke are probably more dangerous than the carbon. In foggy weather the accumulation of these substances in the lower strata may be such as to cause irritation of the eyes, nose, and respiratory passages, leading to asthmatic breathing and bronchitis and, in extreme cases, to death. The Meuse Valley fog disaster will probably become a classic example in the history of gaseous air pollution. Released in a rare combination of atmospheric calm and dense fog, it is believed that sulphur dioxide and other toxic gases from the industrial region of the valley caused 63 sudden deaths, and injuries to several hundred persons. Physical examination showed difficult breathing, rapid pulse, cyanosis, cardiac dilation, and a redness and inflammation of the mucosa of the nose, mouth, throat, trachea, and bronchi.

Carbon monoxide from automobiles and from chimney gases con-

stitutes another important source of aerial pollution in busy cities. During heavy traffic hours and under atmospheric conditions favorable to concentration, the air of congested streets is found to contain enough *CO* to menace the health of those exposed over a period of several hours, particularly if their activities call for deep and rapid breathing. In open air under ordinary conditions the concentration of *CO* in city air is believed to be insufficient to affect the average city dweller or pedestrian.

Occlusion of Solar Radiation

The loss of light, particularly the occlusion of solar ultra-violet light due to smoke and soot, is beginning to be recognized as a health problem in many industrial cities. Measurements of solar radiation in Baltimore[1] by actinic methods show that the ultra-violet light in the country was 50 per cent greater than in the city. In New York City[2] a loss as great as 50 per cent in visible light was found by the photo-electric cell method.

The effect of air pollution on the health of city dwellers is difficult to determine, owing to the slowness of its manifestations. The aesthetic and economic objections to air pollution are so definite, and the effect of air-borne pollen can be shown so readily as the cause of hay fever and other allergic diseases, that means and expenses of prevention or elimination of this pollution have seemed justifiable to the public.

INDUSTRIAL AIR POLLUTION

In many industrial processes, sufficient amounts of dusts, fumes, and vapors are liberated to be injurious to the health of workers. Some dusts are poisonous (lead, mercury, arsenic, manganese, and cadmium) and some act as irritants (silica, steel, iron, and granite). Certain dusts may produce catarrhal conditions and increase susceptibility to such diseases as bronchitis, pneumonia, and tuberculosis. Silicious dust is especially harmful because it has a direct damaging action upon the tissue of the lungs, but organic dusts, both animal and vegetable (hair, pollen, textile, and fiber), do not seem to affect the lungs at all, although they may cause considerable discomfort in the upper respiratory passages to persons sensitive to them.

Industrial gases and fumes act specifically upon the mucous membranes, the lungs, blood, skin, and eyes. Some extremely poisonous gases act after very short exposures. Among these are carbon monoxide, hydrogen sulphide, ammonia, chlorine, bromine, arsine, and cyanogen.

The industrial processes which liberate harmful substances are too manifold and the effects too diverse to be considered here, where discussion is limited to the commonest and most serious with which the ventilating engineer may be confronted: namely, carbon monoxide, lead, and silica. For a more thorough treatise on the subject reference should be made to books by Hamilton[3], Rosenau[4], and Henderson and Haggard[5].

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[1]Effects of Atmospheric Pollution upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz and F. A. Korff (*American Journal of Public Health*, p. 7, Vol. 19, 1929).

[2]Studies in Illumination, by J. E. Ives (*U. S. Public Health Service Bulletin No. 197*, 1930).

[3]Industrial Poisions in the United States, by Alice Hamilton.

[4]Preventive Medicine and Hygiene, by Milton J. Rosenau.

[5]Noxious Gases, by Y. Henderson and H. Haggard.

Carbon Monoxide Poisoning

Carbon monoxide is a common form of poisonous industrial gas, met with in mines, foundries, coke-oven sheds, garages, and houses. Its action is due to the fact that the combining power of carbon monoxide with the haemoglobin of the red blood corpuscles is about 300 times greater than that of oxygen. Since the resulting stable combination destroys the power of the haemoglobin to unite with oxygen in the lungs and to supply it to the tissues, the effects are due to lack of oxygen, and the symptoms are those of anoxemia: namely, dizziness, headaches, sleepiness, fatigue, and, in extreme cases, paralysis and death. The dangerous saturation level of the blood with carbon monoxide is about 50 per cent. Even as little as 0.07 per cent in the air will render, in half an hour, one quarter of the red corpuscles incapable of uniting with oxygen. One to two parts per 10,000 parts of air is set as a safe limit of pollution which may be breathed for a long time without producing perceptible symptoms.

Silicosis

Silicosis is a chronic disease of the lungs which results from the local physio-chemical action of hydrated silica upon the pulmonary tissue, causing progressive lymphatic fibrosis, and rendering the tissue susceptible to tuberculosis. The disease is slow in evolution, requiring usually a number of years of exposure. It occurs principally among granite workers, sand blasters, metal miners, metal polishers, potters, and millstone workers.

Lead Poisoning

Lead posioning is the most insidious and most common of all industrial diseases. It occurs principally among lead workers and smelters, lead miners, potters, painters, typesetters, stereotypers, plumbers, and workers with glass, gold and silver. Lead, in practically all forms, is a cumulative poison which is absorbed by way of the blood stream, chiefly from the respiratory tract, but also from the digestive tract and from the skin. The effect may be either an acute or chronic poisoning. The principal symptoms are colic, constipation, anemia, headache, anorexia, a bluish line along the edges of the gums, rheumantic pains, and, in extreme conditions, paralysis, blindness, insanity, and death.

It has been found[6] that 2 mg per day is the smallest dose, by inhalation, which in the course of years may result in lead poisoning. Regular inhalation during the usual working hours, of air containing less than 0.2 mg of lead per cubic meter does not seem to produce serious lead poisoning in individuals of representative industrial groups[7].

Prevention

The prevention of industrial hazards from dusts and poisonous gases is largely a ventilation problem consisting of keeping the impurities in air down to a safe concentration. As yet there are no generally accepted standards on which to base the design of the ventilation equipment.

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[6]Lead Poisoning, by Thomas Morrison Legge (*Journal Royal Society Arts*, 1929, Vol. 77, p. 1023).

[7]What is a Dangerous Quantity of Lead Dust in Air, by C. M. Salls (*Industrial Hygiene Bulletin*, New York State Department of Labor, 1925).

Approximate data on the toxicity of various gases and fumes met with in industrial establishments are given in Table 1. Column 5, giving the maximum allowable concentrations for prolonged exposures, was compiled from experiments in which most exposures lasted not more than a week, and it is reasonable to assume that over more prolonged exposures such concentrations would cause pernicious effects.

Much is known concerning the physiological and pathological effects induced by various types and concentrations of atmospheric pollutants. In the absence of an accepted standard for safe breathing, and because of the slow, cumulative effects of certain kinds of air contaminants, the best procedure is the periodic medical examination of individuals, and the routine measurement and study of the concentration and the physical and chemical characteristics of the dusts to which those individuals are exposed.

ABATEMENT OF ATMOSPHERIC POLLUTION

Successful abatement of atmospheric pollution requires the combined efforts of the combustion engineer, the public health officer, and the public itself. The complete electrification of industry and railroads, and the separation of industrial and residential communities would aid materially in the effective solution of the problem.

In the large cities where the nuisance from smoke, dust and cinders is the most serious, limited areas obtain some relief by the use of district heating. The boilers in these plants are of large size designed and operated to burn the fuel without smoke, and some of them are equipped with dust catching devices. The gases of combustion are usually discharged at a much higher level than is possible in the case of buildings that operate their own boiler plants.

SMOKE ABATEMENT

In general, time, temperature and turbulence are the essential requirements for smokeless combustion. Anything that can be done to increase any one of these factors will reduce the quantity of smoke discharged. Especial care must be taken in hand-firing bituminous coals. (See Chapter 27).

Checker or alternate firing, in which the fuel is fired alternately on separate parts of the grate, maintains a higher furnace temperature and thereby decreases the amount of smoke.

Coking and firing, in which the fuel is first fired close to the firing door and the coke pushed back into the furnace just before firing again, produces the same effect. The volatiles as they are distilled thus have to pass over the hot fuel bed where they will be burned if they are mixed with sufficient air and are not cooled too quickly by the heat-absorbing surfaces of the boiler.

Steam or compressed air jets, admitted over the fire, create turbulence in the furnace and bring the volatiles of the fuel more quickly into contact with the air required for combustion. These jets are especially helpful for the first few minutes after each firing. *Frequent firings* of small charges shorten the smoking period and reduce the density. *Thinner*

fuel beds on the grate increase the effective combustion space in the furnace, supply more air for combustion, and are sometimes effective in reducing the smoke emitted, but care should be taken that holes are not formed in the fire. A *lower volatile coal* or a higher gravity oil always produces less smoke than a high volatile coal or low gravity oil used in the same furnace and fired in the same manner.

The installation of more modern or better designed fuel burning equipment, or a change in the construction of the furnace, will often reduce

TABLE 1. TOXICITY OF GASES AND FUMES IN PARTS PER 10,000 PARTS OF AIR[a]

VAPOR OR GAS	RAPIDLY FATAL	MAXIMUM CONCENTRATION FOR FROM 1/2 TO 1 HOUR	MAXIMUM CONCENTRATION FOR 1 HOUR	MAXIMUM ALLOWABLE FOR PROLONGED EXPOSURE
Carbon monoxide	40	15–20	10	1
Carbon dioxide	800–1000	-------	-------	-------
Hydrocyanic acid	30	1 1/2	1/2	1/5
Ammonia	50–100	25	3	1
Hydrochloric acid gas	10–20	1/2	-------	1/10
Chlorine	10	1/2	-------	1/100
Hydrofluoric acid gas	2	1/10	-------	1/33
Sulphur dioxide	4–5	1/2–1	-------	1/10
Hydrogen sulphide	10–30	5–7	2–3	1
Carbon bisulphide	-------	11	5	1/2
Phosphene	20	4–6	1–2	-------
Arsine	2 1/2	1/2	1/2	-------
Phosgene	Over 1/4	1/4	-------	1/100
Nitrous fumes	2 1/2–7 1/2	1–1 1/2	-------	1/3
Benzene	190	-------	31–47	1 1/2–3
Toluene and xylene	190	-------	31–47	-------
Aniline	-------	-------	1–1 1/2	1/10
Nitrobenzene	-------	-------	1/100	1/500
Petrol	243	100–220	-------	-------
Carbon tetrachloride	480	240	40	16
Chloroform	250	140	50	2
Tetrachlorethane	73	-------	-------	1 1/2
Trichlorethylene	370	-------	-------	-------
Methyl chloride	1500–3000	200–400	70	5–10
Methyl bromide	200–400	20–40	10	2
Lead vapor	-------	-------	-------	5–6

[a]Original data compiled by Y. Henderson and H. Haggard. (See *Noxious Gases*, 1927). Data revised by T. M. Legge. (See *Lessons Learned from Industrial Gases and Fumes*, Institute of Chemistry of Great Britain and Ireland, London, 1930).

smoke. The installation of a Dutch oven which will increase the furnace volume and raise the furnace temperature, often produces satisfactory results.

In the case of new installations, the problem of smoke abatement can be solved by the selection of the proper fuel-burning equipment and furnace design for the particular fuel to be burned and by the proper operation of that equipment. Constant vigilance is necessary to make certain that the equipment is properly operated. In old installations the solution of the problem presents many difficulties, and a considerable investment in special apparatus is necessary.

Legislative measures at the present time are largely concerned with the

smoke discharged from the chimneys of boiler plants. Practically all of the ordinances limit the number of minutes in any one hour that smoke of a specified density, as measured by comparison with a Ringelmann Chart (Chapter 40), may be discharged.

These ordinances do not cover the smoke discharged at low levels by automobiles, and, although they have been instrumental in reducing the smoke emitted by boiler plants, they have, in many instances, increased the output of chimney dust and cinders due to the use of more excess air and to greater turbulence in the furnaces.

Legislative measures in general have not as yet covered the noxious gases, such as sulphur dioxide and sulphuric acid mist, which are discharged with the gases of combustion. Where high sulphur coals are burned, these sulphur gases present a serious problem.

DUST AND CINDER ABATEMENT

The impurities in the air other than smoke come from so many sources that they are difficult to control. Only those which are produced in large quantities at a comparatively few points, such as the dust, cinders and fly-ash discharged to the atmosphere along with the gases of combustion from burning solid fuel, can be readily controlled.

Dusts and cinders in flue gas may be caught by various devices on the market, such as fabric filters, dust traps, settling chambers, centrifugal separators, electrical precipitators, and gas scrubbers, described in the following paragraphs.

The cinder particles are usually larger in size than the dust particles; they are gray or black in color, and are abrasive. Being of a larger size, the range within which they may annoy is limited.

The dust particles are usually extremely fine; they are light gray or yellow in color, and are not as abrasive as cinder particles. Being extremely fine, they are readily distributed over a large area by air currents.

The nuisance created by the solid particles in the air is dependent on the size and physical characteristics of the individual particles. The difficulty of catching the dust and cinder particles is principally a function of the size and specific gravity of the particles.

Lower rates of combustion per square foot of grate area will reduce the quantity of solid matter discharged from the chimney with the gases of combustion. The burning of coke, coking coal, and sized coal from which the extremely fine coal has been removed will not as a general rule produce as much dust and cinders as will result from the burning of non-coking coals and slack coal when they are burned on a grate.

Modern boiler installations are usually designed for high capacity per square foot of ground area because such designs give the lowest cost of construction per unit of capacity. Designs of this type discharge a large quantity of dust and cinders with the gases of combustion, and if pollution of the atmosphere is to be prevented, some type of catcher must be installed.

Dust and Cinder Catchers[8]

The various types of dust and cinder catchers available today can be divided into six general classes:

1. Settling chambers.
2. Dust and cinder traps.
3. Centrifugal separators.
4. Electrostatic precipitators.
5. Gas scrubbers.
6. Fabric filters.

The selection of the proper type of catcher calls for a careful study of the material to be caught and the draft and space available. After installation, constant vigilance is necessary to keep the catchers in proper working condition if satisfactory operation is to be obtained.

If possible, the dust or cinder catcher should be installed on the inlet side of the induced draft fans because the dust and cinders in the gases seriously erode the wheels of the fans, the inlet connections and the scrolls. Where the induced draft fans operate at high tip speeds and no catchers are installed, it is not uncommon for the fans to require major repairs within one year and complete replacement within five years.

Settling Chambers

Probably the oldest form of dust catcher is the settling chamber, which generally consists of a large-sized, gas-tight space into which the dust-laden gases are discharged before being delivered to the chimney. The velocity of the gas should be reduced to a point where the larger and heavier particles will be precipitated by gravity. For good operation, the velocity of the gas should be reduced to a maximum of 2 fps. The bottoms of the chambers should be provided with dump plates through which the collected dust can be removed. Because these chambers are not effective in removing the finer dust particles they have been practically superseded by smaller and less costly devices.

Traps, Catchers, Precipitators

Various types of traps have been devised. In general they all depend upon breaking the gas up into thin strata and subjecting those thin strata to several abrupt changes in direction. The dust is thrown out of the gas stream into specially shaped pockets, or impinged against a roughened surface. The trapping pockets are drained into a hopper below with a small quantity of gas and the dust settles out by gravity due to the low velocity in the hopper. In the roughened surface type, various sections of the trap are closed off at intervals by means of dampers and the dust is shaken off the roughened surface into a hopper below.

These devices work very well in catching large size dust and cinders and trap much of the fine dust. They have been used most extensively on stoker-fired installations. They have the advantage of low pressure drop, relatively small space requirements, and low first cost.

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[8]See Smoke and Dust Abatement, by M. D. Engle (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Centrifugal catchers obtain separation by projecting the particles tangentially out of the gas stream. The effectiveness of this type of catcher varies directly as the specific weight of the dust and as the square of the tangential velocity, and inversely as the radius of rotation.

Electrostatic precipitators are used for catching fine dust. These precipitators consist of dust-tight chambers in which are suspended reinforced concrete slabs on about 10-in. centers. Between the slabs are suspended bare metal rods. High-voltage unidirectional current is applied to the reinforcing rods in the concrete slabs acting as positive electrodes, the bare rods acting as negative electrodes. The dust-laden gas flows horizontally through the precipitator and the dust particles migrate toward the concrete slabs to which they adhere and then fall or are scraped off into the dust hoppers below.

Gas Scrubbers

Wet scrubbers have been used for many years for removing dust from gases. A number of different types of scrubbers are now being built for removing dust from boiler flue gases. One type depends upon saturating the gas and washing the dust out of suspension by a spray of water. For best results with this type, the water should be atomized into as fine a spray as possible.

Another type depends upon splitting the gas into thin strata and subjecting these strata to a number of abrupt changes in direction, throwing the dust against the wet surfaces. The main problem in developing a satisfactory wet dust catcher is to find suitable materials of construction that will resist the corrosive action of the wash water for a reasonable length of time.

Fabric Filters

Filters of many kinds have been used with variable success. The filter bags are made of cotton, wool or asbestos fabric. The fabrics used in these filters do not withstand the temperatures at which gases are usually discharged from the boilers, and hence the gases must be cooled by some means. Surface coolers or water sprays can be used for reducing the gas temperatures.

One of the serious objections to all of these dust catchers is the relatively high cost of installation and maintenance, and the space required for installation.

Disposal of Dust and Cinders

Even after the dust and cinders have been caught, the disposal of the material caught presents a serious problem. The cinders discharged with the gases from stoker-fired boilers are usually very high in carbon and contain from 50 to 80 per cent as much heat per pound as the coal which is being burned. It is possible, and usually economical, to burn these cinders. They cannot be satisfactorily mixed with the coal in the stoker hopper but they can be blown into the furnace over the stoker fuel bed and burned satisfactorily. If a sufficient quantity of cinders is caught, a small unit pulverizer can be installed to prepare them for burning over the stoker fuel bed. The same pulverizer can be used for coal at times of

peak load and will materially increase the capacity of the fuel-burning equipment for the boiler to which it is connected.

No satisfactory market has been developed for the dust caught from pulverized coal installations, but the possibilities are being investigated and it seems likely that in the future this material will have a market value that will go a long way toward paying the fixed charges on the cost of catching it.

The distribution of dust in the gas entering and leaving the dust and cinder catchers is not uniform and is different in practically every installation, and varies widely with changes in furnace conditions. In order to obtain a representative sample it is necessary to traverse the inlet and outlet of the catcher with a sampling tube which faces into the gas flow. The velocity of the gas into the sampling tube must be the same as the velocity of the gas in the duct at the instant the sample is taken. The swirls and eddy currents in the ducts make it difficult to obtain consistent readings, but if the test is conducted by some one of experience, an indication of the approximate efficiency can be obtained.

Nature's Dust Catcher

Nature has provided means for catching solid particles in the air and depositing them upon the earth. A dust particle forms the nucleus for each rain drop and the rain picks up dust as it falls from the clouds to the earth. In fact, without dust in the air to form the nuclei for rain drops it would never rain, and the earth would be continually enveloped in a cloud of vapor.

Chapter 16

AIR CLEANING EQUIPMENT

Requirements of an Air Cleaner, Types, Air Washers and Scrubbers, Viscous Type Filters, Dry Air Filters, Air Filter Installations

AIR cleaning devices are intended to remove impurities in air brought into a building for ventilating or air conditioning purposes. These impurities include carbon (soot) from the incomplete combustion of fuels burned in furnaces and automobile engines, particles of earth, sand, ash, automobile tires, leather, animal excretion, stone, wood, rust and paper, threads of cotton, wool and silk, bits of animal and vegetable matter, bacteria and pollen. Microscopic examination shows that the character of the impurities varies with the locality, but as a rule carbon forms the greater part of them while the total is somewhat proportional to the state of industrial activity and the wind intensity. Additional information on sources of air pollution will be found in Chapter 15.

Observations have shown that practically all atmospheric impurities are less than 5 microns in size. (One micron equals 0.001 millimeter or approximately 0.00004 in.). The size and composition of each individual particle determines its buoyancy and consequently the length of time it will remain in suspension. The chart, Fig. 1, shows graphically the sizes of impurities found in the air, and other related data.

To estimate the probable dust load for air filter installations, the following approximate averages of atmospheric dust concentration may be used (7000 grains equal 1 lb):

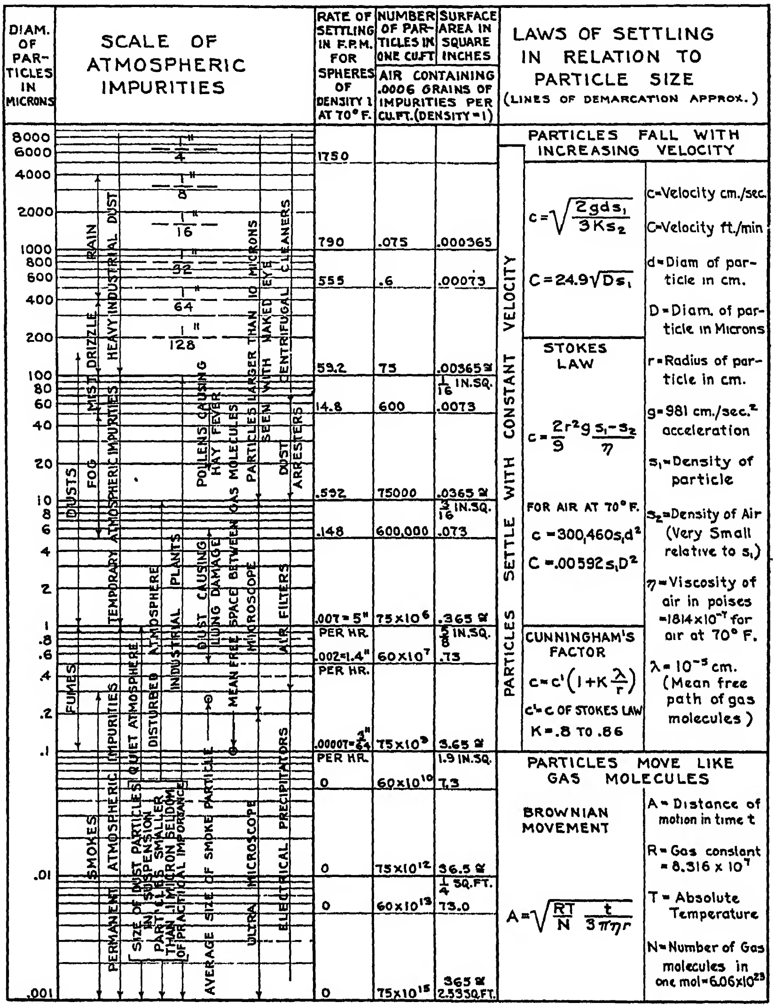
Rural and suburban districts	0.2 to 0.4 grains per 1000 cu ft
Metropolitan districts	0.4 to 0.8 grains per 1000 cu ft
Industrial districts	0.8 to 1.5 grains per 1000 cu ft

REQUIREMENTS OF AN AIR CLEANER

To fulfill the essential requirements of clean air, an air cleaner should:

1. Be efficient in the removal of harmful and objectionable impurities in the air, such as dust, dirt, pollens, bacteria.
2. Be efficient over a considerable range of air velocities.
3. Have a low frictional resistance to air flow, that is, the pressure drop across the filter, measured in inches of water, should be as low as possible.
4. Have a large dust-holding capacity without excessive increase of resistance, or have ability to operate so as to keep the resistance constant automatically.
5. Be easy to clean and handle, or clean itself automatically.
6. Leave the air passing through the cleaner free from entrained moisture or charging liquids used in the cleaner.

The A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilating Work[1] explains how such devices are rated by (1) capacity in cubic feet of air handled per minute, (2) resistance



Compiled by W. G. Frank and Copyrighted.

FIG. 1. SIZES AND CHARACTERISTICS OF AIR-BORNE SOLIDS

in inches of water at rated capacity, (3) dust arrestance, the percentage relationship expressing dust removal efficiency at rated capacity, (4) reconditioning power, the energy necessary to operate the mechanism of

[1]Submitted at the Semi-Annual Meeting of the A.S.H.V.E., June, 1933.

an automatic air cleaning device, and (5) dust holding capacity, the amount by weight of standard dust which a non-automatic air cleaning device will retain before reconditioning is necessary.

TYPES OF AIR CLEANERS

According to the Code, the following four classifications are given the devices:

Class A. Automatic Type: In general all air cleaning devices which use power to automatically recondition the filter medium and maintain a non-varying resistance to air flow.

Class B. Low Resistance Non-Automatic Type: Air cleaning devices for warm air furnaces, unit ventilating machines and similar apparatus and installations in which a maximum of not more than 0.18 in. water gage is available to move air through the air cleaning device.

Class C. Medium Resistance Non-Automatic Type: Air cleaning devices for systems in which a maximum of not more than 0.5 in. water gage is available to move air through the air cleaning device.

Class D. High Resistance Non-Automatic Type: Air cleaning devices for the air intake of compressors, internal combustion engines, and the like, where a pressure of 1.0 in. or more water gage is available to move air through the air cleaning device.

Air cleaners may be also classified as follows:

1. According to principle of air cleaning.
   a. Air washers.
   b. Viscous air filters.
      (1) Unit type.
      (2) Automatic type.
   c. Dry air filters.
2. According to application.
   a. For central fan systems of ventilation and air conditioning. Filters of the automatic or semi-automatic type are usually recommended and are installed in a central plenum chamber.
   b. For unit ventilators. Filters of viscous unit or dry type, installed at inlet of individual units.
   c. For window installations. Self-contained units consisting of fan and filter, usually dry type, adapted to be placed in the ordinary window.
   d. For warm air furnaces. Unit type viscous or dry filters placed in small plenum chamber of warm-air house heating systems.
   e. For compressors and diesel engines. Unit type viscous or dry filters, installed at air intake of compressors and diesel engines.
   f. For compressed air lines. Unit type viscous or dry filters.

With the growing congestion of large cities and an industrial growth throughout the entire country, the percentages of foreign material in the air, such as soot or carbon, which are unaffected by an air washer type of air cleaner, have increased. This has brought about the development of the viscous and dry type air filters which are part of many ventilating and air conditioning systems.

AIR WASHERS AND SCRUBBERS

Information on air washers will be found in Chapter 11.

Scrubbers have not been used very extensively in the past for cleaning

air for ventilating purposes. However, new types have been developed which appear to have possibilities for cases where the air to be cleaned is extremely dirty or where a higher degree of cleanliness is desired than can be obtained with an air washer.

VISCOUS TYPE FILTERS

The principle of air cleaning used in viscous filters is that of *adhesive impingement.* Dust and dirt in the air, especially soot and carbons, are trapped and retained by successive impingements on coated surfaces. While the arrangement of filtering media and the kind of materials used are almost unlimited, there are certain rather definite requirements for a practical commercial filter.

Investigations in this country and abroad demonstrate that the first impingement of dust laden air on a viscous coated surface removes about 60 per cent of the dust, the next impingement takes 60 per cent of what then remains—that is, 24 per cent—and the next impingement removes 9.6 per cent. To secure maximum efficiency, it is necessary to divide the air into innumerable fine streams, as the more intimately and freely the air is brought into contact with the viscous-coated media the better will be the cleaning.

The binding liquid used with viscous filters should have the following properties:

1. Its surface tension should be such as to produce a homogeneous film-like coating on the filter medium.
2. The viscosity should vary only slightly with normal changes of temperature.
3. It should be germicidal in its action to prevent the development of mold spores and bacteria, on the filter media.
4. The liquid should flow freely at low temperatures.
5. Evaporation should not exceed 1 per cent.
6. It should be fireproof.
7. It should be odorless.

Viscous Unit Filters

In the unit type viscous filter, the filtering media are arranged in units of convenient size to facilitate installation, maintenance, and cleaning. Each unit consists of an interchangeable cell or replaceable filter pad and a substantial frame which may be bolted to the frames of other like units to form a partition between the source of dusty air and the fan inlet. The necessary washing, draining, and recharging equipment should be installed near each group of unit filters, with hot water and sewer connections provided.

To secure greater dust holding capacity and a practically constant resistance and air volume, the filter media are usually placed in the direction of air flow, with progressive filter densities determined by the percentage of dust impinged. This arrangement provides relatively large spaces for the collection of dirt in the front of the filter where the bulk of the dust is taken out without undue increase in resistance, while at the back of the filter the openings are smaller to secure high efficiency in the removal of the finer dust particles.

The resistance of a well-designed unit filter of the adhesive impinge-

ment type usually depends upon the velocity at which the air is handled and upon whether the unit is clean or dirty. The cleaning efficiency of the unit is usually highest after it has accumulated a certain portion of its maximum load of dirt because some dust collected in the cell acts as an efficient medium for the further seizing of solids from the air. By periodically cleaning a predetermined number of cells, the resistance and capacity of a built-up filter may be held at any desired figure. The frequency of cleaning any unit filter installation depends upon the dust concentration

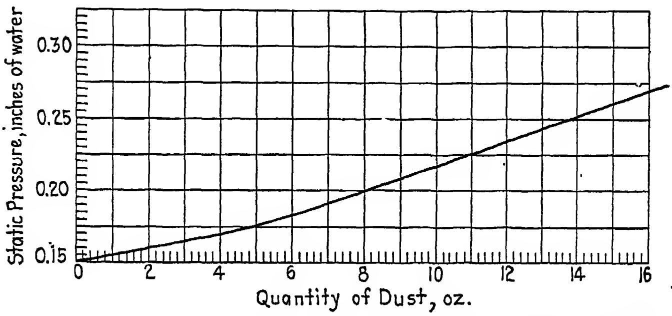


FIG. 2. CHART SHOWING CHANGE IN RESISTANCE DUE TO DUST ACCUMULATION

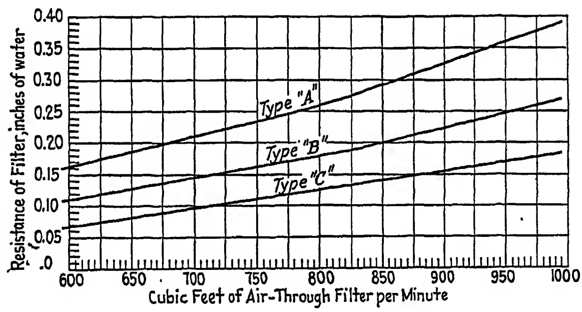


FIG. 3. RESISTANCE TO AIR-FLOW OF A TYPICAL UNIT AIR FILTER

of air being cleaned, and on the amount of dirt which can be accumulated in the filter medium without causing excessive resistance.

Filters consisting of inexpensive frames of cardboard or similar material filled with viscous-coated glass wool or steel wool are available. Because of their construction these units may be discarded when dirty and replaced with new units at relatively little expense. They are used in general ventilation work and with warm air furnaces and other installations where first cost and low resistance to air flow are essential. The operating characteristics of these units conform in general with those of the rigid frame type.

Viscous Automatic Filters

The principle of air cleaning used in the viscous automatic filters is the same as in the unit filters. The removal of the accumulated dust,

however, is done automatically instead of by hand. The automatic cleaning and recoating of these filters is based on the principle that the viscous fluid itself will perform the cleaning function, thereby eliminating a separate washing agent. The dust collected by the filter thus is deposited finally in the bottom of the viscous fluid reservoir from where it may be removed by different methods, depending on the design of the filter.

There are three general types of automatic filters. They are differentiated from each other according to the process of self-cleaning and renewing of the viscous coating used by each type, as follows:

1. The filter medium has the form of an endless curtain suspended vertically, with its lower portion submerged in a viscous fluid reservoir. The curtain rotates slowly through this bath, thus performing the cleaning and recoating of the filter medium.
2. The filter screen is arranged in the form of shelves or cylinders, and the viscous fluid is flushed through all parts of the medium in a direction opposite to the air flow.
3. The filter medium is arranged vertically and is stationary. The viscous fluid is flushed from above over the medium, while the air flow is stopped.

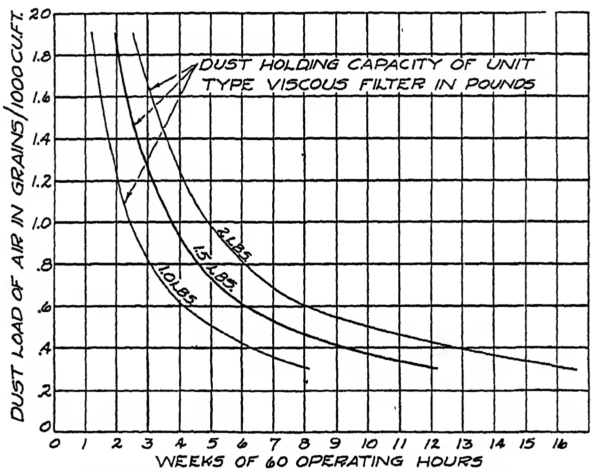


FIG. 4. MAINTENANCE CHART FOR UNIT TYPE VISCOUS FILTERS

The washing and renewing process in automatic filters usually is intermittent. It is accomplished by an electric motor or by other motive power and is controlled by manual or by automatic timing devices. The operating cycle is of a predetermined frequency and should be so timed as to insure a constant static pressure drop across the filter. The customary resistance to air flow is ⅜-in. water gage at an air velocity of 500 fpm, measured at the filter entrance. Automatic viscous filters are made up in units which are delivered either fully assembled or in parts to be assembled at the point of installation.

DRY AIR FILTERS

Dry air filters, in which dust is impinged upon or filtered through screens made of felt, cloth, or cellulose, are available in various types. These filters require no adhesive liquid, but depend on the straining or screening action of the filtering medium. Because of the close texture

of the filtering media used in most of the dry filters, the surface velocity, or velocity of the air entering the media, ranges between 10 and 50 fpm, depending on the nature and texture of the fabric. This necessitates a relatively large screen surface, and the filter media are usually arranged in the form of pockets to bring the frontal area within customary space requirements.

As in viscous unit filters, an average constant resistance and air volume may be obtained by periodic reconditioning or renewal of the filter screens. Since some materials suitable for dry filtering media are affected considerably by moisture which tends to cause a rapid increase in resistance, they should be treated or processed to minimize the effect of changes in humidity.

Filters using felt and similar materials as filter media depend upon vacuum cleaning for reconditioning. A special nozzle, operated from a portable or stationary vacuum cleaner, is shaped to reach all parts of the filter pockets. Permanent filter media should be capable of withstanding repeated vacuum cleanings without loss in dust removal efficiency. While most dry filters are cleaned by replacing an inexpensive filter sheet, the useful life of these sheets often may be lengthened by vibrating or vacuum cleaning.

AIR FILTER INSTALLATIONS

The published performance data for all air filters are based on *straight through* unrestricted air flow. Filters should be installed so that the face area is at right angles to the air flow whenever possible. Eddy currents and dead air spaces should be avoided and air should be distributed uniformly over the entire filter surface, using baffles or diffusers if necessary.

The most important requirements of a satisfactory and efficiently operating air filter installation are:

1. The filter must be of ample size for the amount of air it is expected to handle. An overload of 10 to 15 per cent is regarded as the maximum allowable. When air volume is subject to increase, a larger filter should be installed.
2. The filter must be suited for the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperatures, and maintenance facilities.
3. The filter type should be the most economical for the specific application. The first cost of the installation should be balanced against depreciation as well as expense and convenience of maintenance.

The following recommendations apply to filters and washers installed with central fan systems:

1. Duct connections to and from the filter should change size or shape gradually to insure even air distribution over the entire filter area.
2. Sufficient space should be provided in front as well as behind the filter to make it accessible for inspection and service. A distance of two feet may be regarded as the minimum.
3. Access doors of convenient size should be provided in the sheet metal connections leading to and from the filters.
4. All doors on the clean air side should be lined with felt to prevent infiltration of unclean air. All connections and seams of the sheet metal ducts on the clean air side should be as air-tight as possible.

5. Electric lights should be installed in the chamber in front of and behind the air filter.

6. Air washers should, whenever possible, be installed between the tempering and heating coils to protect them from extreme cold in winter time.

7. Filters installed close to air inlet should be protected from the weather by suitable louvers, in front of which a large mesh wire screen should be provided.

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Chapter 17

FANS

Performance, Fan Efficiency, Characteristic Curves, Selection of Fans, Controls, Designation of Fans, Motive Power, Electric Power

FANS are used for producing air flow except where positive displacement is required, in which case compressors or rotary blowers are used. Fans are classified according to the direction of air flow as (1) *axial flow* or *propeller* type if the flow is parallel with the axis, and (2) *radial flow* or *centrifugal* type if the flow is parallel with the radius of rotation.

Axial flow fans are made with various numbers of blades of a variety of forms. The blades may be of uniform thickness (sheet metal), either flat or cambered, or may be of varying thickness of so-called aerofoil section (airplane propeller type). Where a propeller fan is intended for operation at comparatively high pressures the hub is sometimes enlarged in the form of a disc and the fan is known as a *disc fan.*

Radial flow or *centrifugal fans* include steel plate fans, pressure blowers, cone fans, and the so-called multiblade fans. All the foregoing types have variations which may be obtained by modification of the proportions or change in the curvature and angularity of the blades. The angularity of the blades determines the speed characteristic, the forward curve corresponding to slow speed and the backward curve to high speed operating characteristics.

A wide variation exists in the demands which have to be met by fan installations. A fan may be required to move large quantities of air against little or no resistance or it may be required to move small quantities against high resistances. Between these two extremes innumerable specific requirements must be met. In general, fans of all types can be made to perform the same duty, although mechanical difficulties, noise or lack of efficiency may limit the use to one or another type. The most common field of service for fans of the propeller type is in moving air against moderate resistances requiring a static pressure of less than 1 in. of water, whereas centrifugal fans are more commonly employed for operation at comparatively high pressures.

PERFORMANCE OF FANS

Fans of all types follow certain laws of performance which are useful in determining the effect of changes in the conditions of operation. These laws apply to installations comprising any type of fan, any given piping system and constant air density, and are as follows:

1. The air capacity varies directly as the fan speed.
2. The pressure (static, velocity, and total) varies as the square of the fan speed.
3. The horsepower varies as the cube of the fan speed.

Example 1. A certain fan delivers 12,000 cfm at a static pressure of 1 in. of water when operating at a speed of 400 rpm and requires an input of 4 hp. If in the same installation 15,000 cfm are desired, what will be the speed, static pressure, and horsepower?

$$\text{Speed} = 400 \times \frac{15{,}000}{12{,}000} = 500 \text{ rpm}$$

$$\text{Static pressure} = 1 \times \left(\frac{500}{400}\right)^2 = 1.56 \text{ in.}$$

$$\text{Horsepower} = 4 \times \left(\frac{500}{400}\right)^3 = 7.81 \text{ hp}$$

When the density of the air varies the following laws apply:

4. At constant speed and capacity the pressure and horsepower vary directly as the density.

Example 2. A certain fan delivers 12,000 cfm at 70 F and normal barometric pressure (density 0.07495 lb per cubic foot) at a static pressure of 1 in. of water when operating at 400 rpm, and requires 4 hp. If the air temperature is increased to 200 F (density 0.06018 lb) and the speed of the fan remains the same, what will be the static pressure and horsepower?

$$\text{Static pressure} = 1 \times \frac{0.06018}{0.07495} = 0.80 \text{ in.}$$

$$\text{Horsepower} = 4 \times \frac{0.06018}{0.07495} = 3.20 \text{ hp}$$

5. At constant pressure the speed, capacity and horsepower vary inversely as the square root of the density.

Example 3. If the speed of the fan of Example 2 is increased so as to produce a static pressure of 1 in. of water at the 200 F temperature, what will be the speed, capacity, and horsepower?

$$\text{Speed} = 400 \times \sqrt{\frac{0.07495}{0.06018}} = 446 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \sqrt{\frac{0.07495}{0.06018}} = 13{,}392 \text{ cfm (measured at 200 F)}$$

$$\text{Horsepower} = 4 \times \sqrt{\frac{0.07495}{0.06018}} = 4.46 \text{ hp}$$

6. For a constant weight of air:

(*a*) the speed, capacity, and pressure vary inversely as the density.

(*b*) the horsepower varies inversely as the square of the density.

Example 4. If the speed of the fan of the previous examples is increased so as to deliver the same weight of air at 200 F as at 70 F, what will be the speed, capacity, static pressure, and horsepower?

$$\text{Speed} = 400 \times \frac{0.07495}{0.06018} = 498 \text{ rpm}$$

$$\text{Capacity} = 12{,}000 \times \frac{0.07495}{0.06018} = 14{,}945 \text{ cfm (measured at 200 F)}$$

$$\text{Static pressure} = 1 \times \frac{0.07495}{0.06018} = 1.25 \text{ in.}$$

$$\text{Horsepower} = 4 \times \left(\frac{0.07495}{0.06018}\right)^2 = 6.20 \text{ hp}$$

FAN EFFICIENCY

The efficiency of a fan may be defined as the ratio of the work done in moving the air (air horsepower) to the horsepower input to the fan. The work done in moving the air may be computed on the basis of either the static or the total pressure. When the static pressure is used in the computation it is assumed that this represents the useful pressure and that the velocity pressure is lost in the piping system and in the air which leaves the system. Since in most installations a higher velocity exists at the fan outlet than at the point of delivery into the atmosphere, some of the velocity pressure at the fan outlet may be utilized by conversion to static pressure within the system, but owing to the uncertainty of friction losses which occur at the places where changes in velocity take place, the amount of velocity pressure which is actually utilized is seldom known, and the static pressure alone may best represent the useful pressure.

The efficiency based upon static pressure is known as the static efficiency and may be expressed as follows:

$$\text{Static efficiency}^{1} = \frac{\text{cfm} \times \text{static pressure in inches of water}}{6369 \times \text{horsepower input}}$$

Different fans may develop the same capacity against the same static pressure and with the same power input, and therefore operate at the same static efficiency, while maintaining different outlet velocities. Where a high outlet velocity is desirable or can be utilized effectively, the static efficiency fails to be a satisfactory measurement of the performance. In many applications of propeller fans, air is circulated without encountering resistance and no static pressure is developed. The static efficiency is zero and its calculation is meaningless. Because of such situations where the static efficiency fails to indicate the true performance, many engineers prefer to base the calculation of efficiency upon the total or dynamic pressure. This efficiency is variously known as the total, dynamic, or mechanical efficiency, and may be expressed as follows:

$$\text{Total efficiency} = \frac{\text{cfm} \times \text{total pressure in inches of water}}{6369 \times \text{horsepower input}}$$

CHARACTERISTIC CURVES

In the operation of a fan at a fixed speed the static and total efficiencies vary with any change in the resistance which is imposed. With different designs the peak of efficiency occurs when the fans deliver different percentages of their wide-open capacity. Variations in efficiency accompany variations in pressures and power consumption which are characteristic of the individual designs and which are influenced particularly by the shape and angularity of the blades. Such variations in pressure, power, and efficiency are shown by characteristic curves.

Characteristic curves of fans are determined by tests performed in accordance with the Standard Test Code for Disc and Propeller Fans,

[1]See Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, Edition of 1932.

Centrifugal Fans and Blowers[2] as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers*. The results of tests are plotted in different ways; the abscissae may be the ratio of delivery, assuming full open discharge as 100 per cent, and the ordinates may be static pressure, dynamic pressure, horsepower and efficiency. Pressures may be expressed in per cent of the maximum pressure in the manner shown in the illustrations in this chapter, but in engineering calculations they are sometimes expressed in proportion to the pressures due to the peripheral velocity.

It should be noted that characteristic curves of fan performance are plotted for a constant speed. Some variation in values of efficiency may

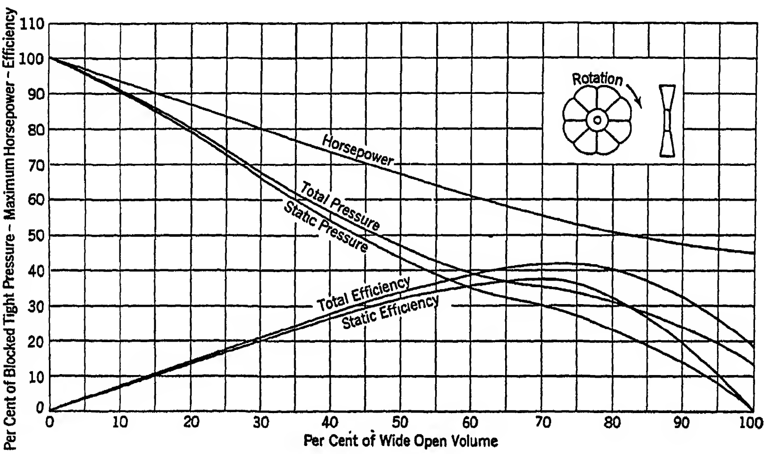


FIG. 1. OPERATING CHARACTERISTICS OF A DISC OR PROPELLER FAN

occur at different speeds but such variation is usually slight within a wide range of speeds. Fans of similar design but of different size will also show some difference in efficiency. The proportions of the housing also affect the performance. As a rule a narrow fan of large diameter shows a higher efficiency than one of greater width and smaller diameter. For a number of designs using blades of certain shapes the proportion of the width to the diameter is so definitely established by the service for which the fan is intended that little variation in efficiency occurs, but in other designs, particularly that which uses straight radial blades, the efficiency may vary over a wide range depending on whether the dimensions are suitable for a fan intended for ordinary ventilating purposes or for a pressure blower. Figs. 1 to 4 show characteristic curves for different types of fans using blades of various shapes, but without reference to the design of housing employed. The efficiency curves are therefore not serviceable for making rigid comparisons of efficiencies obtainable with blades of the various shapes but are intended merely to show reasonable values and

[2]A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923. Amended June, 1931.

more particularly to show the manner in which variations occur with changes in fan capacity.

Propeller fan characteristics are indicated by Figs. 1 and 2. These fans, when properly designed, have a satisfactory efficiency at low resistance, comparing favorably in this respect with centrifugal fans. They are low in cost and economical in operation and occupy relatively little space. Although this type of fan can operate against considerable resistance, the noise often becomes objectionable, so that it does not always compare favorably with centrifugal fans for such service. With most of the designs which employ blades of uniform thickness the power increases rapidly with an increase in resistance.

The curves (Fig. 1) show the rapid reduction in capacity and increase in

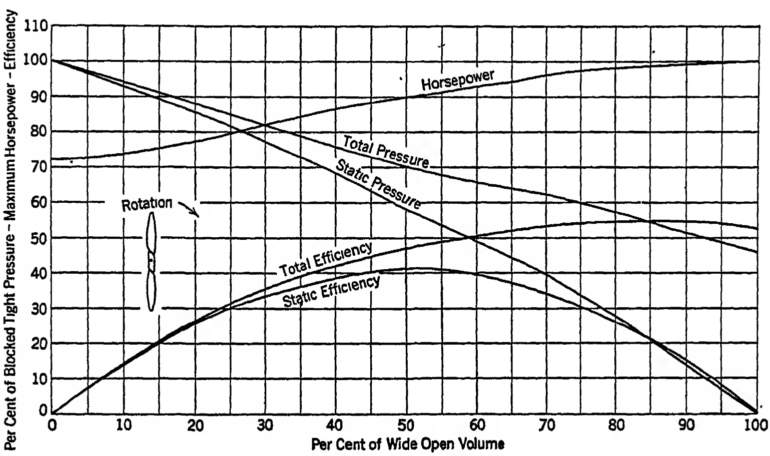


FIG. 2. OPERATING CHARACTERISTICS OF AN AIRPLANE PROPELLER FAN

power as the resistance increases. The low efficiency when overcoming heavy resistance is due to the low speed of the blades near the hub as compared to the relatively high peripheral or tip speed. The air driven by the blade area near the rim can pass back through the less effective blade area at the hub more easily than it can overcome the duct resistance.

Fig. 2 shows the performance of the *airplane propeller fan* in which the blades are similar in shape to those of an airplane propeller but of varying number according to the pressure to be developed. This fan usually operates at a higher speed than does the former type of propeller fan, and with a different power characteristic, the power remaining fairly constant throughout the range of pressures, being somewhat less at the higher than at the lower pressures. The flatness of the pressure curve indicates the advantage of this type of fan in preventing overloading of motors where fluctuations in pressure occur. Variations in the diameter, width, pitch, camber, and the thickness of the blades provide a considerable degree of flexibility in design, so that the peak of total efficiency may be made to occur at wide-open volume or at various percentages of that volume.

The straight blade (paddle-wheel) or partial backward curved blade type

of fan is practically obsolete for ventilation. Its use is largely confined to such applications as conveyors for material, or for gases containing foreign material, fumes and vapors. The open construction and the few large flat blades of these wheels render them resistant to corrosion and prevent material from collecting on the blades. This type of fan has a good efficiency, but the horsepower steadily increases as the static pressure falls off, which requires that the motor be selected with a moderate reserve in horsepower to take care of possible error in calculation of duct resistance.

The *forward curved multiblade fan* is the type most commonly used in heating and ventilating work, as it has a low peripheral speed, a large capacity and is quiet in operation. The point of maximum efficiency for this fan occurs near the point of maximum static pressure. The static

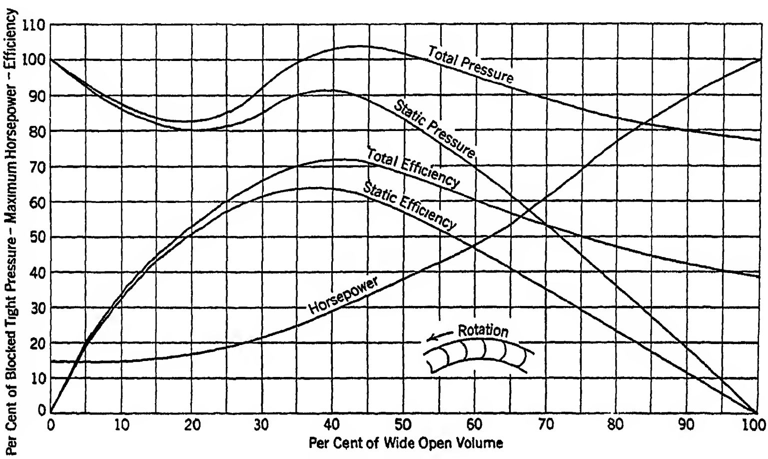


FIG. 3. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED FORWARD

pressure drops consistently from the point of maximum efficiency to full open operation. Fig. 3 shows that this type of fan will have a high and low delivery for a given static pressure at constant speed. The power curve rises continually from low to peak capacity, but if reasonable care is exercised in figuring resistance there is no danger of overloading the motor.

The outstanding characteristics of the *full backward curve multiblade type fan* are the steep pressure curves, the non-overloading power curve, and the high speed. (See Fig. 4). This fan operates at a peripheral speed of approximately 250 per cent of the forward curve multiblade type for like results. The pressure curves begin to drop at very low capacity and continue to fall rapidly to full outlet opening. The steep pressure curves tend to produce constant capacity under changing pressures. Where wide fluctuations occur, the use of this type of fan is desirable to prevent overloading of motors. The maximum power requirement occurs at about the maximum efficiency. Consequently a motor selected to carry the load at this point will be of sufficient capacity to drive the fan over its full range of capacities at a given speed. The high speed of this type makes it

adaptable for direct connected electric motor drives. The high speed necessitates heavier construction, and more operating attention and service is required than for the other type multiblade fans. The dimensional bulk for a given duty often is 150 to 200 per cent that of a forward curve multiblade type fan.

Between the extremes of the forward and the full backward curve blade type fans a number of modified designs exist, differing in the angularity or in the shape of the blades. Common among these designs are the straight radial blade type, the radial tip type, and the double curve blade fan with a forward angle at the heel and a slight backward angle at the tip of the blade. Characteristic curves of these types show varying degrees of

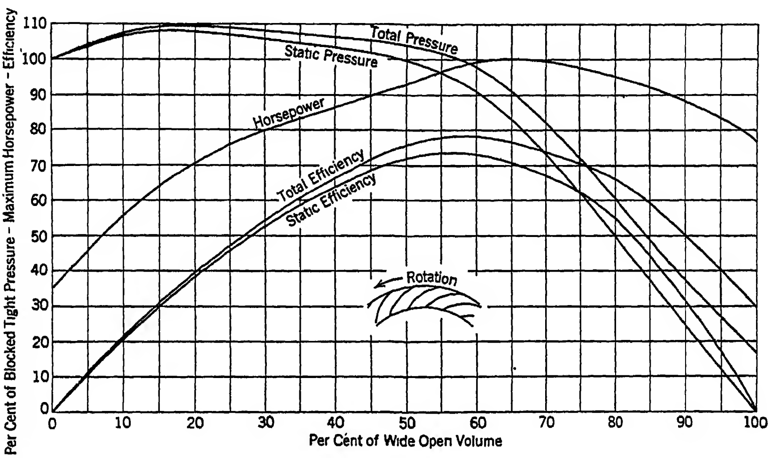


FIG. 4. OPERATING CHARACTERISTICS OF A FAN WITH BLADES CURVED BACKWARD

resemblance to the forward and the backward curve blade characteristics according to the degree of similarity to one or the other of these two designs.

SELECTION OF FANS

The following information is required to select the proper type of fan:

1. Cubic feet of air per minute to be moved.
2. Static pressure required to move the air through the system.
3. Type of motive power available.
4. Whether fans are to operate singly or in parallel on any one duct.
5. What degree of noise is permissible.
6. Nature of the load, such as variable air quantities or pressures.

Knowing the requirements of the system, the main points to be considered for fan selection are (1) efficiency, (2) reliability of operation, (3) size and weight, (4) speed, (5) noise, and (6) cost.

In order to facilitate the choice of apparatus, the various fan manufacturers supply fan tables or curves which usually show the following factors for each size of fan operating against a wide range of static pressures:

1. Volume of air in cubic feet per minute, (68 F, 50 per cent relative humidity, 0.07488 lb per cubic foot).
2. Outlet velocity.
3. Revolutions per minute.
4. Brake horsepower.
5. Tip or peripheral speed.
6. Static pressure.

The most efficient operating point of the fan is usually shown by either bold or italicized figures in the capacity tables.

Fans for Ventilation

Two important factors in selecting fans for ventilating systems are efficiency (which affects the cost of operation) and noise. First cost and space available are secondary. The fans should be selected to operate at maximum efficiency without noise. Noise in a ventilating system is irritating and a cause for complaint. Fans must be selected of proper size in order to reduce it to a minimum. Noise may be caused by other factors than the fan, namely, high velocity in the duct work, unsatisfactory location of the fan room, improper construction of floors and walls and poor installation. Where noise is chargeable directly to the fan, it is caused either by excessive peripheral speeds, or the fan is of insufficient size. It should be remembered, however, that the tip speed required for a specified capacity and pressure varies with the type of blade, and a tip speed that is excessive for the forward curved type is not necessarily so for the backward or slightly backward type. A noisy fan usually is one which is operated at a point considerably beyond maximum efficiency.

For a given static pressure there is a corresponding outlet velocity and peripheral speed wherein maximum efficiency is obtained. If a fan be selected to operate at this point, the cost of operation and the noise can be held within control.

To aid in selecting fans as near as possible to the point of maximum efficiency, there are listed for each static pressure corresponding outlet velocities and tip speeds which will give satisfactory results. The proper tip speed for a given static pressure varies with design of wheel and number of blades or vanes in wheel.

Lower outlet velocities than listed in Table 1 may be used, but care must be exercised when fans of the forward curved type are used to avoid selecting a fan for operation below its useful range.

In exhaust ventilating systems where the air column moves toward the fan, noise due to the higher tip speeds and outlet velocities will not be so readily transmitted back through the air column to the building. Therefore higher outlet velocities may be used, but this will be at the expense of increased horsepower.

Amply large fans should always be used for both exhaust and supply systems, as there may be and usually is leakage despite the most careful workmanship, necessitating the delivery of more air at the fans than is exhausted from or supplied through the openings in the various rooms.

Long runs of distributing ducts, heaters, and air washers usually are parts of any ventilating system where high static pressures are needed.

Table 1. Good Operating Velocities and Tip Speeds for Forward Curved Multiblade Ventilating Fans

Static Pressure In Inches of Water	Outlet Velocity Feet per Minute	Tip Speed Feet per Minute
¼	1000–1100	1520–1700
⅜	1000–1100	1760–1900
½	1000–1200	1970–2150
⅝	1100–1300	2225–2450
¾	1200–1400	2480–2700
⅞	1300–1600	2660–2910
1	1500–1800	2820–3120
1¼	1600–1900	3162–3450
1½	1800–2100	3480–3810
1¾	1900–2200	3760–4205
2	2000–2400	4000–4500
2¼	2200–2600	4250–4740
2½	2300–2600	4475–4970
3	2500–2800	4900–5365

Under such conditions it is practicable to select fans with higher outlet velocities and peripheral speeds since the duct system itself will tend to muffle objectionable air sounds.

The connection of a fan to a metallic duct system should be made by canvas or a similar flexible material so as to prevent the transmission of fan vibration or noises. Where fans are connected to concrete ducts such as are used in the ventilation of vehicular tunnels, the connection is made direct.

Fans for Drying

Both propeller and centrifugal types of fans are used for drying work. Propeller fans are well adapted to the removal of moisture-laden air when operating against low resistance and when handling air at low temperatures. Motors on these fans usually are of the fully-enclosed moisture-proof types so that saturated air or air containing foreign material will not injure the motors.

Unit heaters employing disc or propeller type fans are widely used in the drying field. In drying, disc or propeller fans may be used where not too much duct work is required and where air is to be delivered against pressure, since the noise developed from the high peripheral speed of these fans is not ordinarily objectionable in process work of this nature.

Centrifugal fans or blowers of the multiblade type generally are selected to supply air for drying, as they are capable of delivering large volumes against all pressures.

Belt driven fans usually are to be preferred to direct-connected fans as they make a more flexible and economical unit. Wherever drying is done throughout the year and where air requirements change as the drying conditions change, the drying can be speeded up or reduced through control of the fan capacity. This may be done by changing the fan speed or by varying the outlet area with dampers.

Due to the low speeds of forward curved multiblade or yaddle-wheel type fans, these can be direct-connected to reciprocating steam engines.

and the exhaust steam may be used in the heating apparatus. In selecting engine driven fans for drying processes, where a large quantity of exhaust steam is used in the heaters, a smaller fan and greater power consumption may be used, because power economy is not essential under this condition.

Where static pressure in a dryer varies, and where battery operation is required, the full backward curved fan is preferred. This type is well adapted for direct-connected motors of the higher speeds.

Fans for Dust Collecting and Conveying

The application of fans or exhausters for handling refuse, dust, and fumes generated by machine equipment is covered in Chapter 21. Information is given regarding the methods for determining air quantities, the velocity required for carrying various materials and the method of determining maintained resistance or total static pressure at which the fan is to operate. The selection of a proper size fan or exhauster is at times governed by the future requirements of the plant. In many instances, additional future capacity is anticipated and should be provided for.

Having determined the necessary volume of air and the maintained resistance or static pressure required, the proper size fan may be selected from the fan manufacturers' performance charts or capacity tables. The fan chosen should be the size that will provide the required ultimate quantities with the minimum power consumption.

FAN CONTROL

Some method of volume control of fans usually is desirable. This may be done by varying the peripheral velocity, or by interposing resistance, as by throttling-dampers. Both methods, since they reduce the volume of air, reduce the power required, except in the control of disc fans where an increase in resistance produces an increase in the power required. In many installations adjustments of volume are desirable during varying hours of the day. In others an increased supply of air in summer over that needed for winter is demanded. Experience is required in deciding whether speed-control or damper-control shall be used for specific cases. Where noise is a factor, it may be exceedingly desirable to reduce the speed at times, while on the other hand, any fan which has its normal speed reduced as much as 50 per cent without *change in resistance* will move only 50 per cent of the air.

DESIGNATION OF FANS

Facing the driving side of the fan, blower or blast wheel, if the proper direction of rotation is clockwise, the fan, blower or blast wheel will be designated as *clockwise*. If the proper direction of rotation is counter-clockwise, the designation will be *counter-clockwise*. (The driving side of a single inlet fan is considered to be the side opposite the inlet regardless of the actual location of the drive)[3].

This method of designation will apply to all centrifugal fans, single or double width, and single or double inlet. Do not use the word "hand," but specify "clockwise" or "counter-clockwise."

[3]Recommendations adopted by the *National Association of Fan Manufacturers*.

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as follows:

Bottom Horizontal: If the line of air discharge is horizontal and below the shaft.

Top Horizontal: If the line of air discharge is horizontal and above the shaft.

Up Blast: If the line of air discharge is vertically up.

Down Blast: If the line of air discharge is vertically down.

All intermediate discharges will be indicated as angular discharge as follows:

Either top or bottom angular up discharge or top or bottom angular down discharge, the smallest angle made by the line of air discharge with the horizontal being specified.

In order to prevent misunderstandings, which cause delays and losses, the arrangements of fan drives adopted by the *National Association of Fan Manufacturers* and indicated in Fig. 5 are suggested.

Single inlet, single width fans should be selected wherever possible. If double width, double inlet fans are selected, care must be taken that both inlets have the same free area. If one inlet of a *forward curved blade* type of fan is obstructed more than the other, the fan will not operate properly, as one half of the wheel will deliver more air than the other half. The *backward curved* and *double curved* type with backward tip operate satisfactorily in double or in parallel operation.

MOTIVE POWER

It is no easy matter to predetermine the exact resistance to be encountered by a fan, or having determined this resistance, to insure that no changes in construction or operation shall ensue which may increase air resistance, thus requiring more fan speed and power to deliver the required volume, or which may reduce air resistance, thus causing delivery of more air and a consequent increase of power even at constant speed.

It is recommended, therefore, for centrifugal type fans that the rated power to be supplied shall exceed the rated fan power by a liberal margin, when *forward curved* types are used. When *backward* or *double curved* blade types are used, motors with ratings very close to that of the fan horsepower can be employed.

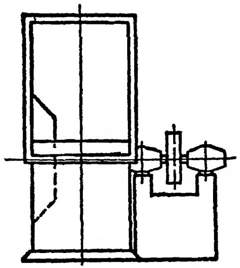
Justification for liberal power provision exists also in the possibility of varying demand due to changes in ventilation requirements, intensity of occupation, and weather conditions.

The motive power of fans should be determined in accordance with the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers, as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers.*

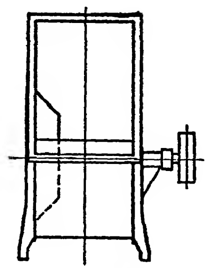
Fans may be driven by electric motors, steam engines (either horizontal or vertical), gasoline or oil engines and turbines, but as previously stated the drive most commonly used is the electric motor.

ELECTRIC POWER

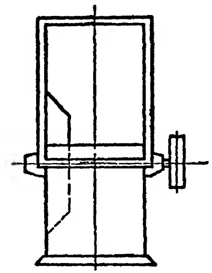
Electric power is almost the universal solution for fan operation, as electric motor speeds are flexible for adaptation to direct-connected fans. Electric motors are readily suited to various types of drives, such as belts, chains or gears.



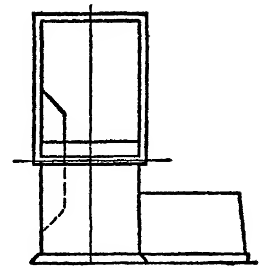
Arr. 1.
For belt drive.
Wheel overhung. Bearing on pedestal.



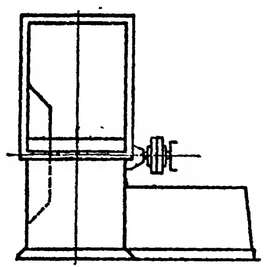
Arr. 2.
For belt drive.
Pulley and wheel overhung. Bearings in bracket on fan housing. Made only in smaller sizes for reversible discharge.



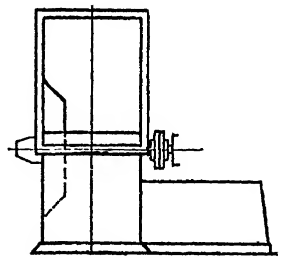
Arr. 3.
For belt drive.
Pulley overhung. Bearings supported on fan housing.



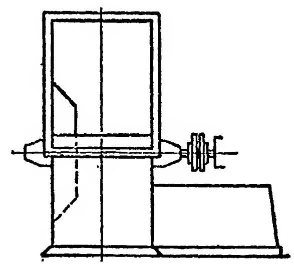
Arr. 4.
For direct drive.
Wheel overhung. No bearings on fan. Wheel mounted on motor or engine shaft. Pedestal for motor or engine.



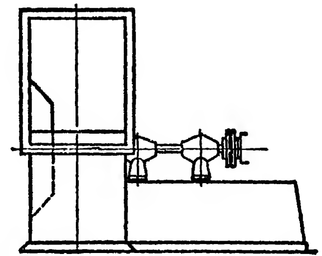
Arr. 5.
For direct drive.
Wheel overhung. Includes housing, wheel, shaft, one intermediate bearing, flanged coupling and pedestal only for motor or engine.



Arr. 6.
For direct drive.
Three-bearing arrangement with fan bearing at inlet side. Includes housing, wheel, shaft, one bearing (in inlet), rigid coupling, and pedestal only for motor or engine.



Arr. 7.
For direct drive.
Similar to Arr. 6, but with two bearings on fan, and flexible instead of rigid coupling.



Arr. 8.
Similar to Arr. 5, but with two bearings on pedestal with motor, and flexible instead of rigid coupling.

FIG. 5. ARRANGEMENT OF FAN DRIVES

Each type of electric motor and kind of electric current has its advantages and disadvantages as applied to a fan.

Direct-connected electric motors usually are very efficient for fan driving because there is no slippage due to belts, and no wear or noise due to chains or gears. There is less maintenance and upkeep to a direct-connected unit, and with an overhung fan wheel on the motor shaft, the usual fan bearings are eliminated.

The disadvantage of a slow-speed direct-connected motor is that it may be unduly large and heavy as well as costly, but this may be offset by the compactness of the unit as a whole due to limited space for fan equipment.

Should anything go wrong with a slow-speed direct-connected motor there may be a considerable delay in securing replacements, as these motors are not usually carried in stock, as is the case with moderately high-speed motors.

If a change of speed is found necessary with a direct-connected motor, it will mean a change of motor, which may necessitate a change in the motor foundation usually built with the fan in such cases. On the other hand, non-direct-connected motors have transmissions subject to wear and slippage, and chains or gears may be noisy with this latter type. However, should a change in speed be necessary where the motor is not direct-connected, changes in speed ratio can easily be accomplished by changing pulleys, sprockets or gears on either the fan or the motor. In the case of a motor breakdown a standard stock motor may easily be substituted.

A type of drive using wedge-shaped rope-like belts, often in multiple, has become very popular recently as it enables the use of high speed motors with slow speed fans. These motors are less expensive and more efficient, and they allow very short belt centers, thus saving floor space and making the fan unit and the motor much more compact than the usual belt drive. The compactness secured by this equipment compares favorably with a direct connected layout. This type of drive is also very quiet in operation, being similar to a conventional belt drive in this respect.

Alternating current motor designs are such that improved operating characteristics are obtained with the higher motor speeds. Efficiencies and power factors are improved over those in effect with slower speed motors, thus showing a considerable saving in power consumption, where some effective speed reducing transmission device to the fan is installed.

Quietness of operation is more readily obtained with moderately high speed induction motors than with low speed motors, as any slight magnetic unbalance is not as easily heard. Magnetic unbalancing at times causes noises whose repetition and wave length is such as to cause vibrations and harmonics. Amplifications of the noises in other parts of a building remote from the motor equipment are sometimes found, due to such noises being carried by the steel work, ducts, or piping in the building. There is considerable evidence that these sounds are more easily controlled with higher motor speeds than with lower motor speeds.

Motors which are practically quiet in operation and free from magnetic disturbing noises can be obtained and should always be specified for

TABLE 2. CLASSIFICATION OF MOTORS

GROUP	SUB-DIV.	TYPE	CURRENT	SPEED CHARACTERISTICS	STARTING TORQUE	STARTING CURRENT	APPLICATIONS
A	1	Shunt Wound	d-c	Constant	Medium	High	Fans
	2	Squirrel Cage	a-c	Constant	Medium	High — About six times full load	Fans, Centrifugal Pumps
	3	Synchronous	a-c	Constant	Medium	Starts as Squirrel Cage Motor	Motor Generator Sets, Air Compressors, Fans
	4	Slip Ring or Wound Rotor	a-c	Constant	Heavy	Low	Vacuum Pumps, Air Compressors
	5	Double Squirrel Cage	a-c	Constant	Heavy	Medium	Frequent and Heavy Starting Loads, Pumps, Compressors
	6	Low-Torque Capacitor	a-c	Constant	Light	Low	Direct-Connected Fans
	7	High-Torque Capacitor	a-c	Constant	Medium	Low	Belt Drive of Fans
	8	High-Torque Capacitor	a-c	Constant	High	Medium	For Heavy Starting Load Such as Larger Fans, Pumps, Compressors
	9	Repulsion Induction	a-c	Constant	High	Medium	Fans, Pumps, Compressors
B	1	Brush Shifting	a-c	Adjustable	Medium	Low	Stokers, Boiler Fans
	2	Cumulative Comp'd with Shunt Predominance	d-c	Adjustable	Heavy	High	Pumps
	3	Squirrel Cage Poles can be Regrouped	a-c	Multi-Speed	Medium	High	Fans, Ice Machines
C	1	Series	d-c	Variable	Heavy	Low	Fans
	2	Cumulative Comp'd with Series Predominance	d-c	Variable	Heavy	Low	Single Acting Reciprocating Pumps
	3	Slip Ring—Using External Resistance in Secondary	a-c	Variable	Heavy	Low	Fans

quietness of operation when used for fan installations in buildings where quietness is a factor.

In the construction of fan and motor foundations where the machinery is mounted on the floor or upon a concrete platform, it is a usual practice to install a layer of cork on top of which is laid or floated the base which carries the apparatus. It is essential that the bolts or lag screws which fasten the machines to this foundation shall not extend through to the floor. It is wise to fasten curbs to the floor, these presenting insulated surfaces to the machinery foundation and so preventing it from traveling. The general classification of motors used for heating, ventilation and air conditioning is shown in Table 2.

Control for Electric Motors

Very small direct current motors may be started by throwing them directly on the line through a suitable starting switch. The larger sizes require some type of starting rheostat. When speed adjustment is desired, the controller for adjusting the speeds of the motor usually functions also as a starting device.

Alternating current motors of 5 hp and under usually may be thrown directly on the line. It is good practice to use a starting switch equipped with a thermal overload or inverse time limit overload device. This type of switch provides protection to the motor beyond the power of fuses to supply. Fuses when used necessarily must be large enough to take care of the inrush current which makes them inadequate for protecting the motor under operating conditions. The thermal overload device allows for this inrush and does not function until an overload has become persistent, the time element depending upon the percentage of overload over the rating of the element. This type of switch is available for manual operation and also is furnished in the magnetic type for remote operation by push button, or for operation by other types of pilots, such as pressure switches and thermostats.

On the standard squirrel cage motors above 5 hp a starting compensator usually is employed to keep the inrush current to within the limits specified by the local power companies. Compensators may be obtained in transformer types and primary resistor types, and usually are furnished for manual operation. They can be secured for remote control also, but are necessarily expensive. However, the new type of high reactance, self-starting motors, may usually be thrown across the line up to 30 hp in size, and still have their inrush current within the limits of the rules of the *National Electric Light Association*. With this type of motor a magnetic contactor usually is used. This device may be operated from a remote point by push button, if desired. These magnetic contactors are furnished usually with thermal overload and no-voltage protection.

For remote operation of motors through magnetic starters, the operating buttons may be located in the engineer's or manager's office, and tell-tale indicating lamps may be wired up with the circuit to indicate whether or not the unit is in operation. This type of control is very desirable in large buildings where the engineer is to have complete charge of the ventilating system.

Remote or automatic control of the units may be affected also by

pneumatic or hydraulic apparatus, or by thermostats or by pressure devices which are provided with electric contacts for starting or stopping the units upon reaching certain conditions.

Variable speed slip ring motors and direct current motors may also be arranged for remote speed control by means of pre-set automatic regulators, where the operating speed of the motor is set by a dial-switch or regulator (which may be near the fan or at a remote point) and the motor is then automatically controlled at this speed merely by operating the remote control push button for starting or stopping the equipment.

Arrangements may be made for remote control of fan motors, or for automatic control by influence of temperature. Remote control may be by pneumatic or by hydraulic manipulation as well as by electrical means.

In many large ventilating systems which have heating plants in connection, steam engines are used to operate fans. A medium speed steam engine, exhausting at low pressure into the radiators which heat the building or which warm the air, is a very economical source of power, is quiet in operation, and has a wide range of speed variation. The steam economy of such an engine usually is of little importance, since the engine serves as an auxiliary to the pressure-reducing valve interposed in such cases between the boiler and the radiators.

Internal combustion engines and line shafting are often used for fan driving, requiring clutches or shift-belts with loose pulleys in order to secure proper starting and control.

It is seldom necessary to reduce the speed of ventilating fans more than 50 per cent from the maximum fan speed, but for economical operation on large systems there should be a number of operating speeds between maximum speed and 50 per cent below maximum speed.

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Chapter 18

SOUND CONTROL

Measurement of Noise, Noise in Buildings, Absorption of Sound, Coefficients of Absorption, Insulation of Air-Borne Sound, Location and Insulation of Equipment Room, Insulation of Machinery and Solid-Borne Vibration, Control of Noise Transmission Through Ducts, Air Currents, Effect of Humidity upon Acoustics

THE part which ventilating and air conditioning plays in the acoustics of buildings becomes apparent from a consideration of the requirements for good hearing in any architectural interior. These requirements are:

1. The room should be free from noise, whether of inside or outside origin.
2. The useful sound, whether speech or music, should be sufficiently loud (with reference to any residual noise) to be heard easily and distinctly.
3. The useful sound should be distributed uniformly in all parts of the room, and the sound reaching the listeners should be free from long-delayed reflections which produce interference or echoes.
4. The room should be free from pronounced resonant tones which may result from either volume or panel resonance.
5. The room should contain sound-absorptive materials in such amounts, and of such qualities, as will provide a proper balance between the persistence and cessation of the articulated components of sound, that is, the reverberation in the room should be long enough to sustain harmony and impart tonal blending to music, and at the same time it must be short enough to prevent the overlapping and confusing of the separate sounds of speech.

Obviously, the first of these requirements is the one which imposes restrictions on the installation of ventilating equipment—the equipment noises must be unobjectionable in occupied rooms—although the fifth requirement is not entirely independent of the humidity and temperature of the air.

LOUDNESS

Loudness is the sensation of sound intensity. When we say one sound is louder than another we imply a difference in intensity level. Two identical whistles when sounded together do not make a sound twice as loud as one. It may take ten to make a sound 20 per cent louder than one. It has been found that loudness bears a logarithmic relationship to intensity of sound. On this basis a scale of loudness has been built and a unit, the decibel (db), has been established. This scale is illustrated in Fig. 1 which shows the loudness of some typical noises. The formula for relating loudness and intensity is:

$$L_1 - L_2 \text{ (db)} = 10 \log_{10} \frac{I_1}{I_2} \qquad (1)$$

where

L = Loudness in db. I = Intensity.

Thus the two whistles made a noise $10 \log_{10} 2 = 3$ db louder than one whistle and the ten whistles $10 \log_{10} 10 = 10$ db louder than one. It would take a hundred whistles to make a noise 20 db louder than one and a thousand to make a noise 30 db louder.

MEASUREMENT OF NOISE

Since the chief acoustical problem in the ventilating or air conditioning of a building consists of reducing equipment noise, it is necessary to describe methods for measuring noise. The measurement of noise is a relatively new problem, and although there are several reliable methods, there are as yet no standardized units, scales, or instruments for measuring noise[1]. However, the *decibel* (*db*) described above is widely used in this country and England as the standard unit for noise or sound intensity—a unit of the same size, but called a *phon*, is used in Germany—and the zero level of the scale is a barely audible sound. Since the relation between subjective loudness and sound intensity is dependent upon pitch, it is customary to refer loudness to a single frequency. A 1000-cycle tone is generally accepted as the reference frequency, that is, the loudness of any sound is rated in terms of an equally loud 1000-cycle tone. Thus, a noise of 50 db means that the noise would be judged to be of the same loudness as a 1000-cycle tone which is 50 db above the normal threshold of audibility for the 1000-cycle tone.

As the frequencies decrease below 1000 cycles, the ear becomes less sensitive, until at about 30-cycles sounds are no longer audible regardless of their intensity. Similarly, for higher frequencies, the limit of audibility is reached around 7000 cycles. Thus, at frequencies below 1000 cycles, sounds of the same loudness must have a greater intensity than at 1000 cycles. This is particularly fortunate, as otherwise the low frequency sounds would mask all others.

Noise measurements are usually made by one of three methods. The first is the electrical instrument method, which uses a noise meter usually consisting of a microphone, an amplifier, and a galvanometer. Where such a meter is to measure the loudness of a noise without regard to the frequency distribution, it must contain a weighted network which electrically simulates the varying sensitivity of response of the ear to different frequencies. Where it is desired to analyze the character of the sound, filters which shut out all but certain bands of frequencies are used with the meter. A number of manufacturers make such meters.

The second method consists essentially of varying the intensity of an artificially generated sound until the noise generated is masked by the noise being measured. Obviously, this method is subject to human errors in observation to which the instrumental method is not, but in the hands of a careful observer quite satisfactory results may be obtained. One instrument used is the audiometer, which consists of a buzzer, an ear phone, and a rheostat. The phone is held a fixed distance from the ear while the resistance of the rheostat is varied until the sound of the buzzer,

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[1]A Committee of the *American Standards Association* is working on this project and it is probable that at least tentative standards will soon be available.

For further information on this subject see, How Sound is Controlled, by V. O. Knudsen (*Heating, Piping and Air Conditioning*, October, 1931) and Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

as transmitted electrically to the phone, can no longer be heard. Audiometers are available either for covering all frequencies, as in the noise meter, or for covering certain frequency bands only.

A third method of measuring noise, simple, yet sufficiently accurate for most field measurements, employs only three tuning forks and a stop watch. Forks having frequencies of 128, 512 and 2048 are recommended. The forks must be calibrated. That is, it is necessary to know for each fork (1) the initial intensity, in number of decibels above its threshold, immediately after it has received a standard hit or excitation, and (2) the damping rate, in decibels per second. These calibrations can be made in any well-equipped acoustical laboratory. A *standard hit or excitation* can be imparted to the fork by a felt-covered spring hammer, or simply by letting the fork fall from a vertical position through an arc of 90 deg, hitting a suitable pad (such as soft rubber or felt for the 128 and 512 forks and hard rubber for the 2048 fork). The average 512 steel fork will have an initial intensity, when held ¼ in. from the ear with the broad side of the prong facing the ear canal, of about 80 db, and will decay at a rate of about 1.0 db per second. Such a fork will remain audible about 80 sec in a perfectly quiet place, provided the listener has normal hearing. In the presence of a noise, it will remain audible until its tone is just *masked* by the noise. Thus, if a 512 fork, having an initial intensity of 80 db and a damping rate of 1.0 db per second, should be found to remain audible 35 sec in the presence of a certain noise, the masking effect of the noise is 80 − 35, or 45 db.

Procedure

The method of measuring any noise is as follows: The observer, in the presence of the noise, strikes the 128 fork a standard blow. At the same instant he starts a stop watch. The fork is then held in front of the ear canal, and moved back and forth slightly, until the tone of the fork is just completely masked by the noise, at which instant the watch is stopped. This measurement is repeated at least two times. The average time is subtracted from the time the 128 fork remains audible in a quiet place. This difference multiplied by the damping rate of the fork gives the masking effect of the noise at 128 cycles. Similar measurements are made with the 512 and 2048 forks. Measurements of this type give a satisfactory description of both the intensity and the frequency distribution of the noise. The average masking effect of the noise at 128, 512 and 2048 cycles will usually be about 5 to 10 db less than the reading given by a noise meter.

NOISE IN BUILDINGS

Measurements of the intensity of speech, music and noise in many buildings, with special consideration of the noise produced by ventilating equipment, have given the results indicated by Fig. 1. The equivalent loudness of sounds in buildings varies from less than 10 db near the outlet of an air duct in a very quiet sound studio to nearly 100 db in a noisy boiler factory. It will be noted that the noise from the ventilating fan in a certain high school auditorium was nearly as loud as average speech in a large auditorium. Such an amount of noise is devastating to good acoustics; in fact, it is impossible to hear speech in the presence of such a noise.

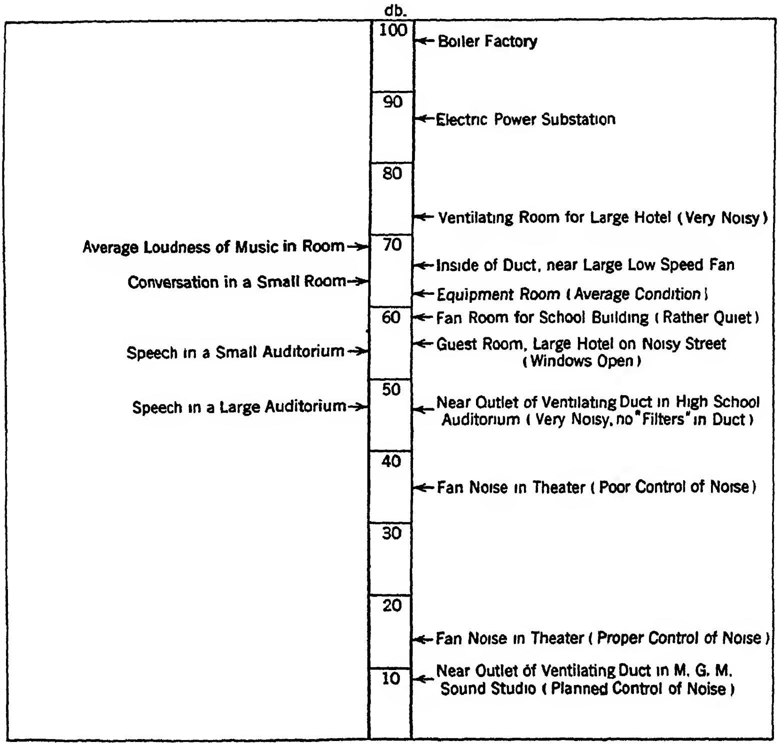

FIG. 1. CHART SHOWING THE EQUIVALENT LOUDNESS (IN DECIBELS) OF SPEECH, MUSIC, AND A NUMBER OF NOISES INCIDENT TO THE VENTILATING OF BUILDINGS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

In every problem of noise reduction in buildings it is necessary to know how much noise can be tolerated. The noise levels given in Table 1 may be regarded as completely inoffensive. They represent what might be termed ideal conditions, not often realized in existing buildings. However, they represent conditions which can be attained by proper control

TABLE 1. ACCEPTABLE NOISE LEVELS

Talking Picture Studios	6 to 8 db
Radio Broadcasting Studios	8 to 10 db
Hospitals	8 to 12 db
Music Studios	10 to 15 db
Apartments, Hotels, Homes, Small Private Offices	10 to 20 db
Theaters, Churches, Auditoriums, Classrooms, Libraries	12 to 24 db
Talking Picture Theaters, Small Clothing Stores	15 to 25 db
General Offices	20 to 30 db
Large Public Offices, Banking Rooms, Upper Stories of Department Stores, Restaurants, Barber Shops	25 to 35 db
Grocery Stores, Drug Stores	30 to 50 db
Accounting and Typewriting Offices	35 to 45 db
Main Floor of Department Stores	40 to 50 db

of noise, and the heating and ventilating engineer should aim to provide the degree of quiet specified in the table.

In considering the tolerable room noise level due to heating, ventilating, or air conditioning apparatus, not only must the absolute value of the noise be considered but also its relation to the room noise level without the apparatus running. This is necessary since a large increase of noise subjects the apparatus to serious criticism even though the level may be low. It must also be borne in mind that the noise produced by the apparatus is additive to that of the room without apparatus. Thus if the two are equal, when combined the noise level will be 3 db higher. For these reasons the room noise caused by the apparatus should not exceed the other room noise.

Noise Control

Essential to the design of a satisfactory system are: *first*, a knowledge of the nature and intensity of the noise generated by the various parts of the equipment; *second*, a knowledge of how to vary the noise level between the apparatus and the conditioned room if need be; *third*, a knowledge of the acceptable level of apparatus noise in the conditioned room. Besides these, the engineer must be able to deal with other noises which might enter the room when openings are made into it, such as *cross talk* between rooms connected with common ducts, and noise transmitted to portions of duct systems outside the conditioned room and thence to its interior.

The problem of apparatus noise is receiving the study of equipment manufacturers who are aiming at both noise reduction and standardization. Some manufacturers now have noise ratings available for their equipment, while some pass each unit of equipment of certain types through sound tests during the course of manufacture.

The problem of noise reduction from apparatus to room must take into consideration and treat separately the three modes of travel of noise to the room: *first*, from the apparatus through the air to the walls of the room and thence to its interior; *second*, through the building structure to the room; *third*, through ducts or openings to the room. Because the noise entering by each of these three channels is susceptible to quantitative analysis, solutions are available. Along with the transmission of sound through the building structure, the engineer must also consider the transmission of vibration, which may also be objectionable. The solution is not complete, however, until the effect of the noise entering the room on the room noise level is determined.

ROOM NOISE LEVEL, COEFFICIENTS OF ABSORPTION

One of the most effective means of reducing noises in ventilating equipment is accomplished by the proper covering of the interior walls and ceiling of the equipment room, or the inner walls of the ducts, with sound-absorptive materials. The intensity I of a continuous sound in a room is

$$I = \frac{E}{a} \text{ or } \frac{I'S'}{a} \qquad (2)$$

where

E = the rate of emission of the noise source = $I'S'$. (The intensities of noises entering the room times the areas through which they enter).

a = the total amount of absorption supplied by the boundaries and contents of the room.

$= \alpha_1 S_1 + \alpha_2 S_2 + \alpha_3 S_3 + \ldots$, *where* S_1, S_2, S_3, ... are the areas of the boundary materials for the room.

α_1, α_2, α_3, ... are the corresponding coefficients of absorption. Hence, by increasing tenfold the absorptivity of the boundaries of a room it is possible to reduce tenfold the average intensity of sound in the room; that is, the intensity level would be reduced 10 db.

Thus it is possible to compute the noise level in the room if the intensity of noises entering the room or generated in it are known.

It will be seen that the noise intensity reduction is dependent upon the amount of sound absorption in the room, and that the first units of absorption are more effective than succeeding units. In general, the room noise level will be from 10 to 20 db lower than the air inlet or outlet noise intensity, the 10 db being in the case of bare rooms having large ventilating or air conditioning openings in relation to their size, and the 20 db in the case of rooms having large amounts of absorptive material with small openings. In some cases, the noise level reduction may run up to as much as 30 db, but then the higher sound intensity adjacent to the openings tends to nullify the effects of the extra reduction. Where these openings are large, the local effect on the noise intensity extends some distance from the opening; for instance, a four square foot opening might

TABLE 2. COEFFICIENTS OF SOUND ABSORPTION[a]

Material	Thickness (Inches)	Coefficients of Sound Absorption 128 Cycles	512 Cycles	2048 Cycles
Acoustex 60, spray painted	1	0.16	0.51	0.72
Acousti-Celotex, Single *B*	⅝	0.11	0.45	0.68
Acousti-Celotex, Triple *B*	1¼	0.20	0.75	0.67
Acoustic Flexfelt	....	0.27	0.56	0.68
Acoustone	1	....	0.66	0.69
Akoustolith plaster	½	0.21	0.29	0.37
Akoustolith *A*, Tile	1	0.14	0.48	0.83
Brick wall, unpainted	18	0.024	0.031	0.049
Calicel	1	0.23	0.72	0.71
Corkoustic, Type C	1½	0.08	0.61	0.64
Glass	....	0.035	0.027	0.020
Insulite Acoustile, Type 44	1⅜	0.26	0.50	0.61
Kalite, with three coats lacquer	¾	0.35	0.43	0.45
Macoustic Plaster, stippled to depth of ½ in.	½	0.13	0.31	0.58
Masonite	7/16	0.18	0.32	0.33
Plaster, gypsum on hollow tile	....	0.013	0.020	0.040
Plaster, gypsum, scratch and brown coats on metal lath on wood studs	....	0.020	0.040	0.058
Plaster, lime, sand finish, on metal lath	¾	0.038	0.060	0.043
Poured concrete, unpainted	....	0.010	0.016	0.023
Rockoustile	1	0.18	0.57	0.72
Sabinite	½	....	0.34	0.49
Sanacoustic Tile	1¼	0.19	0.79	0.74
Stuccoustic Plaster, Type XB	¾	0.29	0.59	0.72
Transite Tile	1	0.19	0.81	0.72
Trutone Tile	1⅛	0.31	0.57	0.64
Wood sheathing, pine	¾	0.098	0.10	0.082
Wood, varnished	....	0.05	0.03	0.03

[a]Architectural Acoustics, by V. O. Knudsen, pp. 219, 220, 240-251.

have a local effect within ten feet, while a one-half square foot opening would have a local effect within only five feet.

The coefficients of sound-absorption for a number of standard absorptive materials used, or suitable for use, in equipment rooms are given in Table 2. Coefficients are given for frequencies of 128, 512 and 2048 cycles. Where the frequency of the noise is not known, the values for 512 or 128 cycles are usually used.

INSULATION OF AIR-BORNE SOUND

The transmission of air-borne sounds through rigid partitions is accomplished primarily by the diaphragm-like vibrations of the partition. The weight per square foot of the wall is the determining factor, and the insulation value of a wall, in terms of the transmission loss in decibels, is proportional to the logarithm of the weight per square foot. Other factors, such as size, stiffness, composition, manner of mounting, and the use of multiple structures separated by air spaces of flexible connectors, contribute to the effective insulation. If the coefficients of sound transmission of different types of structures and the noise intensity in the space adjoining a room are known, it is possible to calculate the noise intensity in a room by the use of formula (1) and the following formulas:

$$I' = I''\tau \tag{3}$$

where

I'' = Noise intensity in space adjacent to room.
τ = Coefficient of sound transmission.

Coefficients of sound transmission for some common walls are shown in Table 3.

Example 1. Suppose the brick wall between an equipment room and an adjacent auditorium has an area of 200 sq ft and a coefficient of sound of 0.00001 (see Table 3); that the auditorium contains 2000 sabines[2] of absorption; and that the noise level in the equipment room is 70 db above zero level.

$$70 - 0 = 10 \log_{10} \frac{I''}{I_0} \text{ (from Formula 1)}$$

$$\frac{I''}{I_0} = 10^7$$

$$\frac{I'}{I_0} = 10^7 \times 0.00001 = 100 \text{ (from Formula 3)}$$

$$\frac{I}{I_0} = 100 \times \frac{200}{2000} = 10 \text{ (from Formula 2)}$$

Room loudness $= 10 \log_{10} 10 = 10$ db

If the sound absorption in the auditorium had been as small as 200 sabines, the sound intensity in the auditorium would have been 10 times as great and the noise level in the auditorium would have been 20 db.

If the rest of the auditorium has an area of 20,000 sq ft with a surrounding noise intensity of 50 db ($I'' = 10^5$) the noise level due to all of the noise entering through the wall would be found as follows

$$\frac{I'}{I_0} = 10^5 \times 0.00001 = 1$$

[2]A *sabine* is 1 sq ft of totally absorptive surface.

$$\frac{I}{I_0} = 10 \text{ (Through equipment wall)} + 1 \times \frac{20{,}000}{2000} = 20$$

Room loudness $\times$ 10 $\log_{10} 20 = 13$ db

Now suppose that there is also a duct having 20 sq ft outlet connecting the room with apparatus having a noise level of 70 db ($I'' = 10^7$) and suppose that there is an assumed attenuation in the duct equivalent to a transmission factor of 0.0002. Then

$$\frac{I'}{I_0} = 10^7 \times 0.0002 = 2 \times 10^3$$

$$\frac{I}{I_0} = 20 \text{ (from above)} + 2 \times 10^3 \times \frac{20}{2000} = 40$$

Room loudness = 10 $\log_{10} 40 = 16$ db

It may be seen how the energies of noises entering a room are added to obtain the final room noise intensity.

The average coefficients of sound transmission (128 to 4096 cycles) for a number of wall and of floor and ceiling partitions are listed in Table 3.

TABLE 3. AVERAGE COEFFICIENTS OF SOUND TRANSMISSION FOR BUILDING PARTITIONS[a]

Description of Partition	Average Coefficient
Brick panel, Mississippi, 8 in.; plastered both sides gypsum brown coat, smooth white finish; good workmanship	0.000010
Brick wall, 2½ in. plaster both sides	0.000032
Brick wall, 2½ in., 2 in. furring strips, ½ in. rigid insulation lath plastered both sides	0.0000016
Brick wall, 4 in., 2 in. furring strips and ½ in. rigid insulation lath, plaster, on one side; other side plastered directly on brick	0.0000040
Concrete flat slab floor construction, re-enforced; floating floor consisting of nailing strips, rough and finish flooring; ½ in. rigid insulation furred out and applied as ceiling	0.0000020
Glass, plate ¼ in.	0.0010
Glass, plate ¼ in. double glazed, 1½ in. separation	0.0001
Metal lath, double, on 1½ in. channels, ¾ in. gypsum plaster; without cross bracing clips; 4 in., connected at edges only	0.000016
Tile, hollow clay partition, three cells, 4 in. x 12 in. x 12 in., wood furring strips, ½ in. rigid insulation, gypsum brown coat, smooth white finish	0.0000050
Wood joists, lower side plastered on wood lath; floating floor consisting of nailing strips, rough and finish flooring	0.0000050
Wood studs, four-paper plaster board, three-coat smooth finish gypsum plaster	0.000010
Wood studs, two ½-in. sheets rigid insulation both sides, joints filled, gypsum scratch and brown coats, smooth white finish	0.000013
Wood studs, 2 in. x 4 in., staggered, metal lath, ½ in. gypsum plaster; 7½ in.; connected at edges only	0.000040

[a]*Architectural Acoustics*, by V. O. Knudsen, pp. 308-322.

LOCATION AND INSULATION OF EQUIPMENT ROOM

The equipment room, if possible, should be located at a considerable distance from all rooms in which quiet is required. If this is not possible, it is necessary to provide a high degree of insulation against the noise which may be transmitted through the walls of the equipment room, and also against the noise which almost certainly will be communicated through the short ducts. (See discussion of Control of Noise Transmission through Ducts, p. 253). Three wall sections and two floor and

ceiling sections which are satisfactory for the wall insulation of the equipment room are shown in Fig. 2. Other partitions, with their sound insulating values, are listed in Table 3. The addition of absorptive materials (such as are described in Table 2) to the inner walls and ceiling of the equipment room will not only increase the insulation through the walls, but will also reduce the intensity of the noise in the room. The equipment room noise intensity may be figured in the same way as that of the conditioned space, taking the equipment as the source of noise. In case the equipment is subject to considerable vibration it is advisable to provide a separate or *floated* floor.

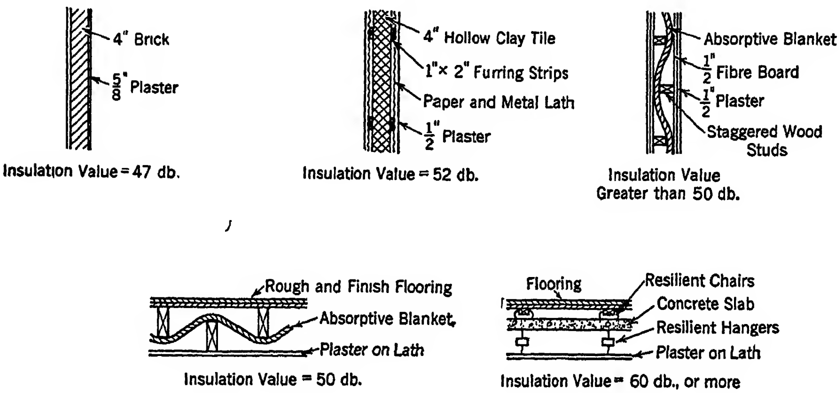

FIG. 2. THREE WALL SECTIONS AND TWO FLOOR AND CEILING SECTIONS WHICH ARE SUITABLE FOR THE INSULATION OF EQUIPMENT ROOMS[a]

[a]Acoustical Problems in the Heating and Ventilating of Buildings, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

INSULATION OF MACHINERY AND SOLID-BORNE VIBRATION

Since mechanical vibrations are readily transmitted through the solid structure of a building, it is extremely important in air conditioning that all mechanical equipment in which vibrations are generated be thoroughly insulated from the solid structure of the building. An almost universal notion prevails that the vibrations generated by machinery can be insulated from a building simply by placing a slab of cork or a layer of hair felt between the machinery and the floor of the room. If the machinery is sufficiently heavy, and the cork or felt sufficiently resilient, this expedient may suffice. On the other hand, if the machinery is not sufficiently heavy to *load* the cork or felt support to the extent that the natural frequency of the machinery on the cork or felt is low in comparison with the frequency generated by the equipment, the cork or felt may be of little avail. The insulation of vibration can be accomplished by means of suitable elastic supports or suspensions, but the design of these elastic supports should be based upon calculation rather than guess-work.

The theory of the insulation of vibration was first worked out by

Soderberg[3]. If a machine of mass m be supported by an elastic pad the amount of vibratory force communicated by the machine to the floor or foundation upon which it rests will be determined by the elastic and viscous properties of the pad. The ratio of the vibratory force communicated to the floor or foundation with the machine resting upon the pad, and with the machine resting directly upon the floor, is given by the following equation:

$$\tau' = \sqrt{\frac{r^2 + \frac{1}{4\pi^2 n^2 c^2}}{r^2 + \left(2\pi nm - \frac{1}{2\pi nc}\right)^2}} \tag{4}$$

where

τ' = the so-called *transmissibility* of the support.
c = the compliance (that is, the reciprocal of the force constant).
r = the mechanical resistance owing to the viscous forces within the support.
n = the frequency of vibration generated by the machine which is to be insulated, such as the commutation frequency of a motor or the blade frequency of a fan.
m = the mass of the machine to be insulated.

It should be noted that not only must vibrations within the audible range of frequencies be considered, but those in the sub-audible range as well, since these may cause objectional vibrations. All the possible frequencies should be considered in the calculation. Sometimes beat effects are introduced by slight irregularities of belts or pulleys that have much lower frequencies than those of the rotating elements.

If the pad is to be of any value in the prevention of solid-borne vibrations, the value of τ' must be considerably smaller than unity. If the fundamental frequency of vibration generated by the machine happens to coincide with the natural frequency of the mass of the machine resting on the elastic pad, a condition of resonance will be established, and the machine will exert a greater force upon the foundation than it would if the pad were completely removed. It is necessary, therefore, that the elastic support be sufficiently compliant, and the mass of the machine sufficiently heavy, that the natural frequency of the mass m upon its elastic support will be low in comparison with the frequencies which are generated by the machine. Thus, if the principal vibrations in the machine be of the order of 100 vibrations per second, the natural frequency of the machine mounted on its elastic support should not exceed about 20 vibrations per second.

If a slab of insulating material be placed under the entire foundation of a machine, as is often done in practice, it may happen that the natural frequency of the machine on its elastic support will be nearly the same as the frequencies which are to be insulated, in which case the elastic support will be worse than nothing. In general, as Equation 4 shows, both m and c should be as large as possible if the vibrations of the machine are to be effectively insulated from the solid structure of the building. Furthermore, the machine should rest upon a rigid floor so that the elastic yielding of the floor is prevented from communicating the machinery vibrations to the solid structure of the building.

The elastic support under the machine acts as a low-pass filter which passes all frequencies below about two times the natural frequency of the machine mounted on its elastic support, but prevents all frequencies

[3]C. R. Soderberg, *The Electric Journal* (January, 1924), and succeeding articles. See also V. O. Knudsen, *Physical Review*, Vol. 32, 1928, p. 324, and A. L. Kimball, Journal *Acoustical Society of America*, Vol 2, 1930, p. 297.

above about $\sqrt{\frac{mc}{\pi}}$ from reaching the solid structure of the building. The principal influence of the internal mechanical resistance r is to limit the vibration at the resonant frequency. It is generally advisable, therefore, to use materials which have an appreciable internal resistance.

The values of c and r can be determined for any specimen of flexible material and, when known, can be used to determine the insulation value of any particular set-up. The value of c can be obtained by making static measurements of the amount of displacement of the compressed support for each additional unit of the compressing force. If this be done for a specimen of the flexible material of a certain thickness and area of cross section, the compliance can be determined for any other thickness or area from the relation that c will be directly proportional to the thickness and inversely proportional to the area of the flexible support. When the internal resistance r is not too large, it can be determined by observing the successive amplitudes of the free vibrations of a mass m which rests upon a specimen of the flexible material, and solving for r by the usual log-decrement method. Or, if the damping be so great that the free motion of m is non-oscillatory, r can be obtained from measurements on the experimentally-determined resonance curve of the forced vibrations of m, or from measurements of the rate of return of m when it is given an initial displacement.

If the resistance of a certain specimen of material, as cork, felt, or rubber, has been determined by any of these methods, the resistance for any other thickness or area of the material can be determined approximately because the resistance will be inversely proportional to the thickness and directly proportional to the area of cross section of the flexible support. Thus, if the values of c and r for a flexible material be known, it is possible to calculate, by means of Equation 4, the amount of insulation that will be obtained from the use of this material as a flexible support for a piece of equipment having a mass m. For the routine calculations in practice, r may be neglected with only a slight sacrifice of accuracy. Table 4 gives the values of c and r for a number of commonly used flexible materials.

In general, there are two principal points to observe in the design of a flexible support for any piece of equipment, namely, the material should have a relatively large compliance and *it should be loaded to nearly the upper safe limit of loading.* Several flexible metallic supports have recently been developed.

Example 2. A machine weighing 1000 lb has a base area of 20 sq ft. Assume that the principal vibration of the machine has a frequency of 100 cycles per second (most machinery vibrations are less than 150 vibrations per second, and the assumed frequency of 100 is quite representative of typical machines). Suppose that a 1-in. slab of corkboard weighing 1.10 lb per board foot be placed between the machine and the floor. The loading on the cork will then be only 50 lb per square foot, or slightly more than $\frac{1}{3}$ lb per square inch. (It is assumed that the compliance c in centimeters per dyne for a specimen 1 in. thick and 1 sq cm in cross-section is 0.25×10^{-6} and the resistance r in mechanical ohms is 0.15×10^{5}).

The *transmissibility* is calculated in the following manner:

Mass of machine in grams $= 1000 \times 454 = 4.54 \times 10^{5}$.

Area of base in square centimeters $= 20 \times 144 \times 2.54 \times 2.54 = 1.86 \times 10^{4}$.

Therefore, the compliance of the entire support, 1 in. thick and 20 sq ft in cross section, is $0.25 \times 10^{-6} \times \frac{1}{1.86 \times 10^4} = 0.134 \times 10^{-10}$ cm per dyne, and the resistance of the entire support is $0.15 \times 10^5 \times 1.86 \times 10^4 = 0.28 \times 10^9$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.28 \times 10^9)^2 + \dfrac{10^{20}}{4\pi^2 \times 100 \times 0.134}}{(0.28 + 10^9)^2 + \left(2\pi \times 100 \times 4.54 \times 10^5 - \dfrac{10^{10}}{2\pi \times 100 \times 0.134}\right)^2}} = 0.93$$

Consequently, it is seen that the *transmissibility* is nearly equal to unity, and that the support therefore is not satisfactory for insulating 100 or fewer vibrations per second.

If the amount of cork be reduced so that it is loaded to 10 lb per square inch, the total area of the supporting cork will be only 100 sq in. or 645 sq cm. The compliance of the entire support will now be $0.25 \times 10^{-6} \times \frac{1}{645} = 0.39 \times 10^{-9}$ cm per dyne, and the resistance will be $0.15 \times 10^5 \times 645 = 0.97 \times 10^7$ mechanical ohms (or absolute units). Therefore

$$\tau' = \sqrt{\frac{(0.97 \times 10^7)^2 + \dfrac{10^{18}}{4\pi^2 \times 100 \times 0.39}}{(0.97 \times 10^7)^2 + \left(2\pi \times 100 \times 4.54 \times 10^5 - \dfrac{10^9}{2\pi \times 100 \times 0.39}\right)^2}} = 0.037$$

It is seen, therefore, that with the bearing surface on the cork reduced to 100 sq in. (that is, with the cork loaded to 10 lb per square inch), the

TABLE 4. COMPLIANCE AND RESISTANCE DATA FOR TYPICAL SPECIMENS OF FLEXIBLE MATERIALS[a]

The compliances and resistances given in the table are for specimens 1 in. thick and 1 sq cm in cross section

MATERIAL	DESCRIPTION OF MATERIAL	APPROXIMATE UPPER SAFE LOADING IN POUNDS PER SQUARE INCH	COMPLIANCE c IN CENTIMETERS PER DYNE	RESISTANCE r IN ABSOLUTE UNITS
Corkboard	1.10 lb per board foot	12	0.25×10^{-6}	0.15×10^5
Corkboard	0.70 lb per board foot	8	0.50×10^{-6}	0.25×10^5
Flax-li-num	1.35 lb per board foot	4 to 6	0.60×10^{-6}	0.50×10^5
Celotex	Carpet lining	10	0.40×10^{-6}	....
Celotex	Insulating board	12	0.18×10^{-6}	....
Insulite	Insulating board	15	0.16×10^{-6}	....
Masonite	Insulating board	15	0.12×10^{-6}	....
Anti-Vibro-Block	....	5	0.60×10^{-6}	1.5×10^5
Sponge Rubber	25 lb per cubic foot	1 to 3	3.0×10^{-6}	....
Soft India Rubber	55 lb per cubic foot	3 to 6	1.2×10^{-6}	....
Hairfelt	10 lb per cubic foot	1 to 2	1.5×10^{-6}	....

[a]*Architectural Acoustics*, by V. O. Knudsen, p. 278.

transmissibility is reduced to 0.037, or the amplitude of vibration transmitted to the floor will be only about 1/27 of what it would be if the machine were mounted directly upon the floor. These two numerical examples will serve to show not only the manner of making the calculations, but also the importance of selecting the proper type and design of flexible supports for insulating the vibrations of a machine from the rigid structure of a building.

CONTROL OF NOISE TRANSMISSION THROUGH DUCTS

The most troublesome sources of noise from ventilating and air conditioning equipment are fan and motor noises which are transmitted through the ducts. The reduction, in decibels, of noise transmitted through a duct, neglecting reflection from ends and bends, is proportional (1) directly to the length of the duct, (2) directly to the perimeter of the duct, (3) inversely to the area of cross section of the duct, and (4) directly (or at least approximately so) to the coefficient of sound absorption of the material which comprises the interior surface of the duct. It is apparent therefore that long narrow ducts, lined with highly absorptive material, will provide a high degree of insulation against the transmission of noise through ducts. In fact, small ducts (4 in. x 6 in.), made of material having a coefficient of sound-absorption of 0.50, will provide a noise reduction of slightly more than 1 db per linear foot.

As can be seen from an inspection of Table 2, noises of low frequency are difficult to absorb; on the other hand, these frequencies are easily reflected by elbows, branches, and duct ends whereas higher frequencies are little affected. Furthermore, the reflection effects are more pronounced in small ducts than in large ducts. Hence, by introducing into a duct a sufficient length of small, absorptive channels together with a number of elbows or other reflecting elements it is possible to reduce the transmitted noise to any required degree. This applies not only to ducts between the equipment room and other rooms in a building, but also to ducts connecting adjacent or nearly adjacent rooms. By the proper use of such *filters* it is possible to eliminate all of the difficulties which arise in connection with the transmission of sound through ventilating ducts. The problem is an engineering one which can be worked out prior to the installing of the equipment, and it can be calculated in such a way as to meet the most rigorous demands for silent operation. There is a need for quantitative data regarding the attenuation or noise-reduction provided by different types of ducts, but even with the meager data available it is possible to design filters which will suppress the ordinary noises incident to the ventilating or air conditioning of buildings[4].

In general, the motion of air resulting from the ventilating of rooms is not sufficient to introduce any appreciable difficulty in auditoriums, except where noise may originate from the issuing of high-speed air from nozzles. However, by proper stream-lining of the nozzles, it is possible to work with speeds which are adequate for all practical purposes without producing any disturbing noises. Since sound is propagated with a velocity of more than 1100 fps, the velocity of the air would have to attain speeds

[4]How Sound is Controlled, by V. O. Knudsen (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

of at least 20 to 30 fps before these wind velocities would have any appreciable influence upon the propagation of sound.

If there is to be any appreciable motion of air in an auditorium, it is advantageous to have the upper layers of air moving in a direction from the stage toward the audience, as this will tend to refract the sound waves down toward the audience. However, unless the speed of the air is as great as 20 or 30 fps, the amount of refraction will not be noticeable. Therefore, as a rule the motion of air in an auditorium does not have an appreciable effect upon the acoustical properties of the room.

EFFECT OF HUMIDITY UPON ACOUSTICS

Recent experiments[5] have shown that both the humidity and the temperature of air have a marked influence upon the rate of absorption of high-pitched sounds. Perfectly dry air is less absorptive than air containing any amount of water vapor. At relative humidities of 5 to 25 per cent, the air is highly absorptive but becomes less and less absorptive as the humidity is increased. High-frequency sounds are propagated better in cold humid air than in hot dry air, and since high-frequency sounds are particularly important for the preservation of good quality in speech and music it is advantageous to maintain the air in a room at a relatively high humidity, not less than about 55 to 60 per cent. On the other hand, where it is desirable to absorb all frequency components of sound, as for the reduction of noise in offices, it is advantageous to maintain relatively dry air.

The time of reverberation in a room is given by the following equation:

$$t = \frac{0.049V}{-S \log_e (1 - \alpha) + 4mV} \tag{5}$$

where

V = volume of room in cubic feet.

S = interior surface of room.

α = average coefficient of sound-absorption of the interior surface of the room.

m = the absorption coefficient of the air in the room.

The coefficient m depends upon the frequency of the sound and the humidity (and probably the temperature) of the air. At a temperature of 70 F, and for sound waves having a frequency of 4096 vibrations per second, $m = 0.0027$ at 25 per cent relative humidity, 0.0018 at 54 per cent, and 0.0013 at 82 per cent. It will be seen, therefore, that the absorption of sound in the air is twice as great at a relative humidity of 25 per cent as it is at a relative humidity of 82 per cent. (This explains why sounds in the open travel so much better on humid days than they do on dry days). Although this dependence of absorption upon humidity is characteristic of low-frequency as well as high-frequency sound, the actual amount of absorption in the air is negligible for frequencies below about 1024 vibrations per second. However, the absorption of the higher frequencies in the air is a significant factor, and its dependence upon humidity calls for careful consideration in planning the air-conditioning equipment for buildings.

[5]Effect of Humidity upon the Absorption of Sound in a Room, by V. O. Knudsen (*Journal Acoustical Society of America*, July, 1931). Also see report presented at the May, 1933, meeting of A. S. of A.

Chapter 19

AIR DUCT DESIGN

Pressure Losses, Friction Losses, Friction Loss Chart, Proportioning the Losses, Sizes of Ducts, General Rules, Procedure for Duct Design, Air Velocities, Proportioning the Size for Friction, Main Trunk Ducts with Branches for Public Buildings, Equal Friction Method, Details of Duct Construction

THE flow of air due to large pressure differences is most accurately stated by thermodynamic formulae for air discharge under conditions of adiabatic flow, but such formulae are complicated, and the error occasioned by the assumption that the gas density remains constant throughout the flow may be considered negligible when only such pressure differences are involved as occur in ordinary heating and ventilating practice.

In the development of the formulae, diagrams and tables for the flow of air, use is made of the following basic formula for the flow of liquids:

$$V = 1096.5\sqrt{\frac{p}{W}} \tag{1}$$

where

V = velocity in feet per minute.
p = velocity head or pressure in inches of water.
W = weight of air in pounds per cubic foot.

For standard air (70 F and 29.92 barometer) $W = 0.07495$ lb per cubic foot. Substituting this value in Equation 1:

$$V = 1096.5\sqrt{\frac{p}{0.07495}} = 4005\sqrt{p} \tag{2}$$

PRESSURE LOSSES

The drop in pressure in air distributing systems is due to the *dynamic* losses and the *friction* losses. The friction losses are those due to the friction of the air against the sides of the duct. The dynamic losses are those due to the change in the direction or in the velocity of air flow.

Dynamic Losses

Dynamic losses occur principally at the entrance to the piping, in the elbows, and wherever a change in velocity occurs. The entrance loss is the difference between the actual pressure required to produce flow and the pressure corresponding to the flow produced; it may vary from 0.1 to

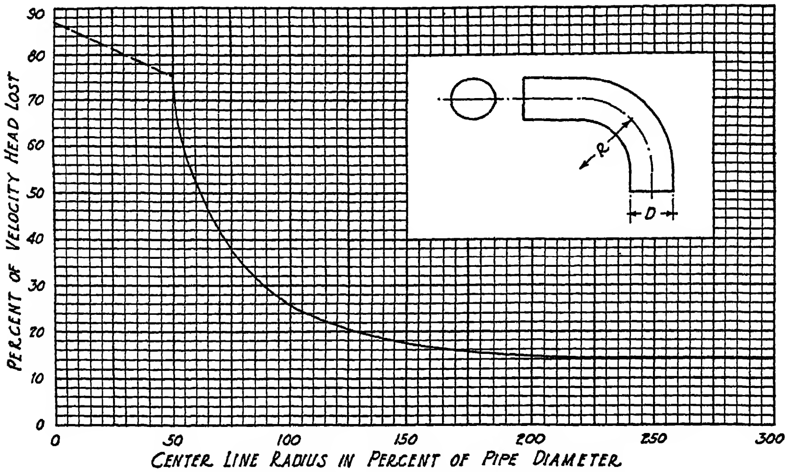

FIG. 1. CURVE SHOWING LOSS OF PRESSURE IN ROUND ELBOWS

0.5 times the velocity head. The pressure loss in elbows must also be allowed for in the design. It is customary to express dynamic losses in terms of the percentage of the velocity head; in other words, the percentage of that pressure corresponding to the average velocity in the duct which is expressed in terms of inches of water gage. Figs. 1 and 2 show the effect of changing the radius of elbows of square and rectangular section. These charts are based on tests of pipe elbows of ordinary good sheet metal construction. For example, a five-piece round pipe elbow having a centerline radius of one diameter has a loss of about 25 per cent of the velocity head. At a velocity of 2000 fpm the corresponding head

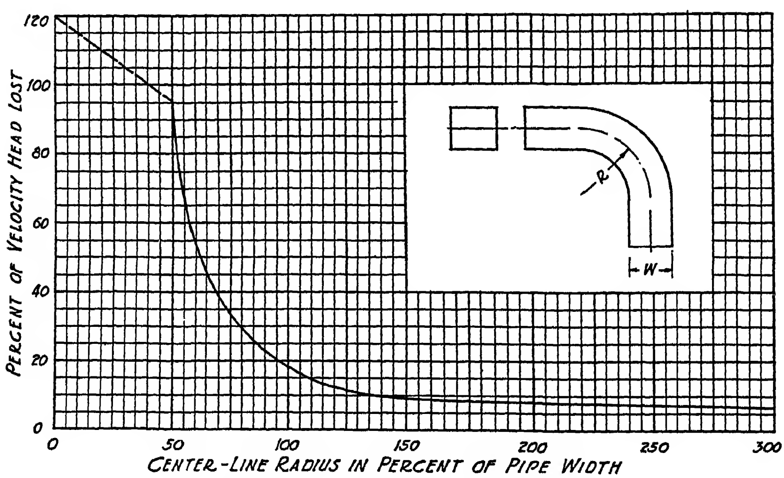

FIG. 2. CURVE SHOWING LOSS OF PRESSURE IN SQUARE ELBOWS

is 0.25 in. water gage, and at this velocity the elbow just referred to would cause a pressure drop of 0.063 in. water gage. Experience has shown that good results may be obtained when the radius to the center of the elbow is 1½ times the pipe diameter. The pressure drop will then be approximately 17 per cent of the velocity head for round ducts, and 9 per cent for square ducts. Very little advantage is gained in making elbows with a radius of more than two diameters.

Friction Losses

Friction losses vary directly as the length of the duct, directly as the square of the velocity, and inversely as the diameter. Since length is a fixed quantity for any system, the factors subject to modification are the area and the velocity, which determine the relation between the first cost of the duct system and the cost of the power for overcoming friction.

The friction between the moving air and pipe surface causes a loss of head which is numerically equal to the pressure required to maintain a given velocity, and is expressed in the following modification of Fanning's formula:

For round pipe and standard air (70 F and 29.92 in. barometer)

$$h_L = f\frac{L}{D}h_V = \frac{L}{CD}\left(\frac{V}{4005}\right)^2 \qquad (3)$$

For rectangular ducts

$$h_L = fL\left(\frac{a+b}{2ab}\right)h_V = \frac{L}{C}\left(\frac{a+b}{2ab}\right)\left(\frac{V}{4005}\right)^2 \qquad (4)$$

where

h_L = loss of head, inches of water.

$h_V = \left(\frac{V}{4005}\right)^2$ = velocity head, inches of water.

V = velocity of air, feet per minute.

L = length of pipe. } all in feet

D = diameter of pipe. } all in feet

a, b = sides of rectangular duct. } all in feet

f = coefficient of friction.

$C = \frac{1}{f}$ = length of pipe in diameters for one head loss.

For all practical purposes C varies only with the nature of the pipe surface: $C = 60$ for perfectly smooth pipe; = 55 for pipe as used in planning mill exhaust systems; = 50 for heating and ventilating ducts; = 45 for smooth and 40 for rough conduits of tile, brick or concrete. However, Fritzche states (and numerous tests check very closely) that f varies inversely as the 2/7 power of the pipe diameter, and inversely as the 1/7 power of the velocity, or inversely as the 1/7 power of capacity, which is the same thing. Thus Formula 3 may be revised as follows, based upon a loss of one velocity head (at 2000 fpm) in a length equal to 50 diameters of 24 in. galvanized swedged pipe:

$$h_L = 1.1\frac{L}{CD^{9/7}}\left(\frac{V}{4005}\right)^{13/7} \qquad (5)$$

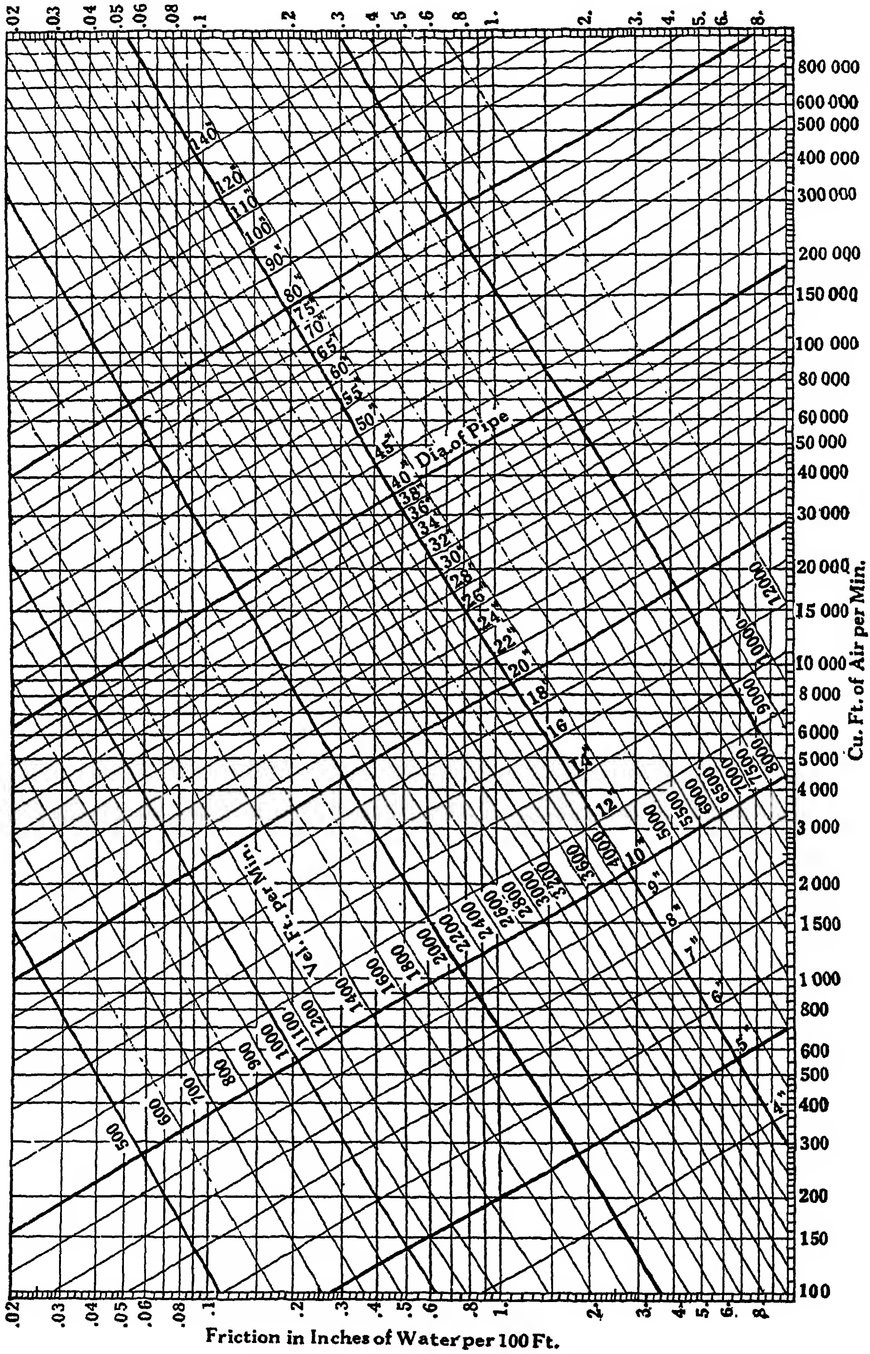

Fig. 3. Friction of Air in Pipes

The proceding formulae are based on standard air, and for other conditions the friction varies directly as the air density and inversely (approximately) as the absolute temperature. The increase of friction due to increase of air viscosity with increased temperature is small and is generally neglected.

Friction Loss Chart

Fig. 3 is a convenient chart for determining the friction loss for various air quantities in ducts of different sizes. The general form of this chart is familiar, but it should be noted that it is corrected for changes in the coefficient of friction based on the rule that the coefficient of friction varies inversely as the 2/7 power of the diameter, and inversely as the 1/7 power of the velocity. Fig. 3 is based on a loss of one velocity head (at a velocity of 2000 fpm) in a length equal to 50 diameters of 24-in. round galvanized-iron duct of the usual construction. Although this chart is laid out for a value of C equivalent to 50, it may be used for other values of C by varying the friction inversely as this constant. For example, if a rougher pipe is used with 40 as the value of C, the friction loss as read from the chart should be multiplied by $\frac{50}{40}$.

Example 1. Assume that it is desired to pass 10,000 cfm of air through 75 ft of 24-in. diameter pipe. Find 10,000 cfm on the right scale of Fig. 3 and move horizontally left to the diagonal line marked 24-in. The other intersecting diagonal shows that the velocity in the pipe is 3200 fpm. Directly below the intersection it is found that the friction per 100 ft is 0.59 in.; then for 75 ft the friction will be $0.75 \times 0.59 = 0.44$ in. In a like manner any two variables may be determined by the intersection of the lines representing the other two variables.

Proportioning the Losses

Other losses of pressure are at the entrance to the duct, through the heating units, air washer, etc. In ordinary practice in ventilation work it is usual to keep the sum of the duct losses ⅓ to ½ and the loss through the heating units at less than ½ of the static pressure. The remainder is then available for producing velocity. In the design of an ideal duct system, all factors should be taken into consideration and the air velocities proportioned so that the resistance will be practically equal in all ducts regardless of length.

SIZES OF DUCTS

The sizes of ducts and flues for gravity or mechanical circulation of air are usually based on the losses due to friction, and these losses must be kept within the available pressure difference. This pressure difference in mechanical ventilation is that derived from the fan, while in gravity ventilation the aspirating effect due to the temperature and height of the column of heated air causes the pressure difference.

General Rules

The general rules to be followed in the design of a duct system are:

1. The air should be conveyed as directly as possible at reasonable velocities to obtain the results desired with greatest economy of power, material and space.

2. Sharp elbows and bends should be avoided.

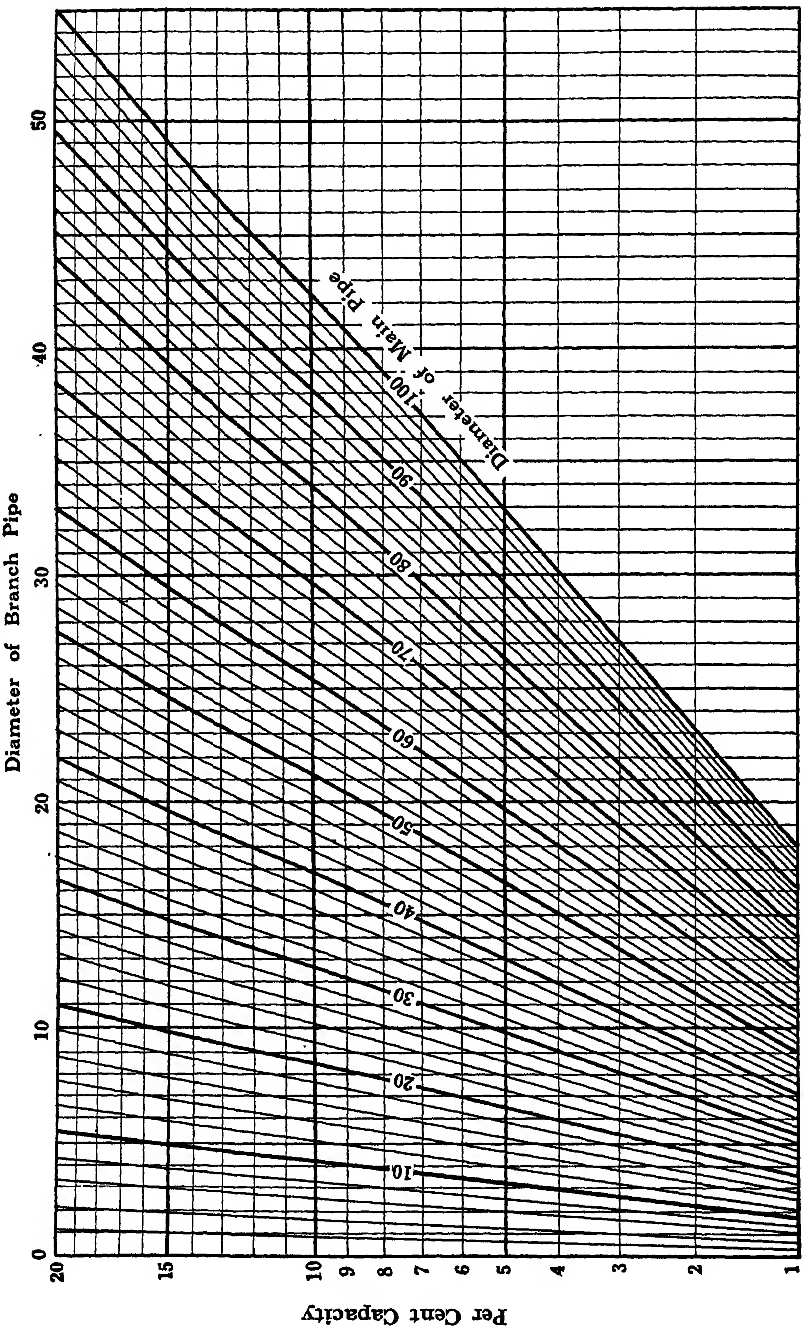

FIG. 4. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH (1 TO 20 PER CENT CAPACITY)

3. The sides of all ducts or flues should be as nearly equal as possible. (In no case should the ratio between long and short sides be greater than 10 to 1).

Procedure for Duct Design

The general procedure for designing a duct system is as follows:

1. Study the plan of the building and draw in roughly the most convenient system of ducts, taking cognizance of the building construction, avoiding all obstructions in steel work, equipment, etc., and at the same time maintaining a simple design.
2. Arrange the positions of duct outlets to insure the proper distribution of heat.
3. Divide the building into zones and proportion the volume of air necessary to supply the heat for each zone.
4. Determine the size of each outlet, based on the volume as obtained in the preceding paragraph, for the proper outlet velocity.
5. Calculate the sizes of all main and branch ducts by either of the following two methods:

   a. Velocity Method. Arbitrarily fix the velocity in the various sections, reducing the velocity from the point of leaving the fan to the point of discharge to the room. In this case the pressure loss of each section of the duct is calculated separately and the total loss found by adding together the losses of the various sections.

   b. Friction Pressure Loss Method. Proportion the duct for equal friction pressure loss per foot of length.

6. Calculate the friction for the duct offering the greatest resistance to the flow of air, which resistance represents the static pressure which must be maintained in the fan outlet or in the plenum space to insure distribution of air in the duct system. The duct having the greatest resistance will usually be that having the longest run, although not necessarily so.

Air Velocities

The following velocities of air are considered standard for public buildings:

1. Through the outside air intakes, 1000 fpm.
2. Through connections to and from heating unit, 1000 to 1200 fpm.
3. Through the main discharge duct, from 1200 to 1600 fpm.
4. In branch ducts, 600 to 1000 and in vertical flues 400 to 800 fpm.
5. In registers or grilles, 200 to 400 fpm depending upon the size and location. If diffusers of proper design are used, 25 per cent higher air velocities are permissible.

These duct velocities may safely be increased 20 per cent if first-class construction is used to prevent any breathing, buckling, or vibration. High velocities at one point in the system neutralize the effect of proper design at all other points; hence the importance of splitters in elbows and similar precautions. For industrial buildings noise is seldom considered, and main duct velocities as high as 2800 or 3000 fpm may be used where conditions will permit. For department stores and similar buildings, maximum velocities with good construction and design may be as high as 2000 or 2200 fpm in main ducts, with suitable reduction in branches and outlets. With these velocities first-class duct construction is essential.

Proportioning the Size for Friction

By means of Figs. 4 and 5 the diameter of branch pipes necessary to carry a given percentage of the total air in the main pipe with the same friction per foot of the length may be determined. These charts, as well as Fig. 3, are based on the assumption that the coefficient of friction varies inversely as the 1/7 power of the capacity.

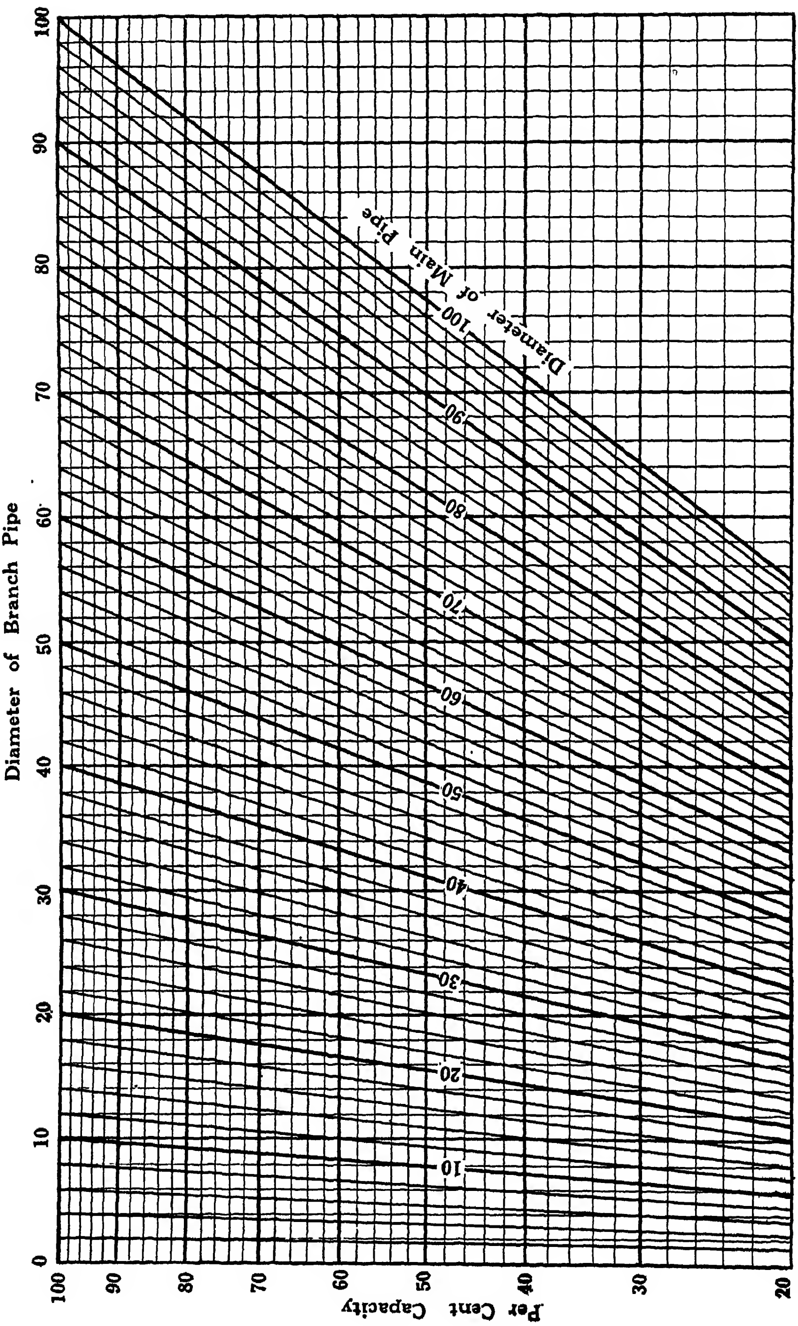

FIG. 5. MAIN AND BRANCH PIPES FOR EQUAL FRICTION PER FOOT OF LENGTH
(20 TO 100 PER CENT CAPACITY)

Example 2. Suppose a 60-in. main pipe is to be used, and it is desired to know the size of branch pipe required to carry 50 per cent of the total air in the main. Find 50 per cent at the left of the chart, move right to the 60-in. diagonal line and note directly above at the top of the chart that the branch pipe will be 46.5 in. in diameter.

Where rectangular ducts are used it is frequently desirable to know the equivalent diameter of round pipe to carry the same capacity and have the same friction per foot of length. Table 1 gives directly the circular equivalent of rectangular ducts for equal friction and capacity. To obtain the size of rectangular ducts for different capacities, but of the same friction per foot of length, first obtain the equivalent round pipe for equal friction. Thus, if a branch of sufficient size to carry 30 per cent of a 12 x 36-in. pipe is desired, it is found from Table 1 that the main is equivalent to a 22.2 in. diameter round pipe. From Fig. 5, 30 per cent of this is a pipe 14.3 in. in diameter, and referring again to Table 1, the rectangular equivalent branch is a 12 x 14-in., 10 x 17¼-in., or any other desirable combination.

Multiplying or dividing the length of each side of a pipe by a constant is the same as multiplying or dividing the equivalent round size by the same constant. Thus, if the circular equivalent of an 80 x 24 in. duct is required, it will be just twice that of a 40 x 12 in. duct, or $2 \times 23.3 = 46.6$ in.

DUCTS FOR PUBLIC BUILDINGS

A main duct with branches is generally used to convey tempered air for ventilation purposes only. In place of individual ducts, a comparatively large main duct supplies air by branches to the room or rooms. The velocities vary according to the nature of the installation and the degree of quietness required. At the start of the run a velocity as high as 2000 fpm may be used, but this is considered the maximum for public building work, and is reduced to from 400 to 800 fpm in the risers. This duct system may be designed so that the loss of pressure in the branches is equalized in a manner similar to that previously described.

Example of Equal Friction Method

Example 3. Fig. 6 shows a typical layout of an air distribution system which is applicable for ventilation of hotel dining rooms, offices, etc.

The volume of air in cubic feet per minute for the room is determined on the basis of the number of air changes per hour required. In the example shown, the room ventilated is a hotel dining room 135 ft x 85 ft x 15 ft. A 7½-minute air change (8 air changes per hour) is assumed for proper ventilation, giving 22,935 cfm as the air required.

The clear area of the fresh air inlet is based on a velocity of 1000 fpm or $\frac{22,935}{1000} =$ 22.94 sq ft. If the air washer is provided with automatic humidity control, the tempering coil should raise the temperature of the entering air to 32 F. The washer with its automatic control will then raise the temperature from 32 F to 42 F. If the washer is not provided with automatic humidity control, the tempering coil must raise the temperature of the entering air to at least 55 F to allow for some temperature drop in the washer due to evaporation. The reheating coil is selected to raise the temperature of the air from that leaving the air washer to 70 F. The air washer should have a maximum velocity of 500 fpm through the clear area, which, in this case, is 46 sq ft. For more detailed information on tempering coil and air washer control, see Chapters 23 and 14.

Since the plan shows a moderately short run of main duct with no risers near the fan outlet, a fan should be selected which will have the required capacity of 22,935 cfm with a maximum velocity through the fan outlet of 1400 fpm. The outlet area, therefore, should be 16½ sq ft.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION*

Side Rectangular Duct	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	24
8	6.1	6.9	7.6	8.2	8.8															
9	6.5	7.3	9.0	8.7	9.3	9.9														
10	6.8	7.7	8.4	9.2	9.8	10.4	11.0													
11	7.1	8.0	8.8	9.6	10.2	10.9	11.5	12.1												
12	7.4	8.3	9.2	10.0	10.7	11.4	12.0	12.6	13.2											
13	7.6	8.7	9.6	10.4	11.1	11.8	12.5	13.1	13.7	14.3										
14	7.9	8.9	9.9	10.8	11.5	12.3	12.9	13.6	14.3	14.9	15.4									
15	8.2	9.2	10.2	11.1	11.9	12.7	13.4	14.1	14.7	15.3	16.0	16.5								
16	8.4	9.5	10.5	11.4	12.3	13.1	13.8	14.5	15.2	15.8	16.5	17.1	17.6							
17	8.6	9.8	10.8	11.8	12.6	13.5	14.2	15.0	15.7	16.3	17.0	17.6	18.2	18.7						
18	8.9	10.0	11.1	12.1	13.0	13.8	14.6	15.4	16.1	16.8	17.4	18.1	18.7	19.2	19.8					
19	9.1	10.3	11.4	12.4	13.3	14.2	15.0	15.8	16.5	17.2	17.9	18.6	19.2	19.8	20.4	20.9				
20	9.3	10.5	11.6	12.7	13.6	14.5	15.4	16.2	17.0	17.6	18.4	19.0	19.7	20.3	20.9	21.5	22.0			
22	9.7	11.0	12.1	13.2	14.2	15.2	16.1	16.9	17.8	18.5	19.2	19.9	20.6	21.3	21.9	22.5	23.1	23.6	24.2	
24	10.0	11.4	12.6	13.8	14.8	15.8	16.8	17.6	18.5	19.3	20.0	20.8	21.5	22.2	22.8	23.5	24.0	24.7	25.2	26.4
26	10.4	11.8	13.1	14.3	15.4	16.4	17.3	18.3	19.2	20.0	20.8	21.6	22.3	23.0	23.8	24.4	25.1	25.7	26.3	27.5
28	10.8	12.2	13.5	14.8	15.9	17.0	18.0	19.0	19.8	20.7	21.5	22.4	23.1	23.9	24.6	25.3	26.0	26.6	27.3	28.5
30	11.0	12.6	13.9	15.2	16.4	17.5	18.5	19.5	20.5	21.4	22.2	23.1	23.9	24.7	25.4	26.2	26.8	27.5	28.2	29.5
32	11.3	12.9	14.3	15.6	16.9	18.0	19.1	20.1	21.1	22.0	22.9	23.8	24.6	25.4	26.2	27.0	27.7	28.4	29.1	30.5
34	11.6	13.2	14.7	16.1	17.3	18.5	19.6	20.7	21.6	22.6	23.5	24.4	25.3	26.2	26.9	27.7	28.5	29.2	30.0	31.3
36	11.9	13.6	15.1	16.4	17.7	19.0	20.1	21.2	22.2	23.2	24.2	25.1	26.0	26.8	27.7	28.5	29.3	30.0	30.8	32.2
38	12.2	13.9	15.4	16.8	18.2	19.4	20.6	21.7	22.8	23.8	24.8	25.8	26.7	27.5	28.4	29.2	30.0	30.8	31.5	33.1
40	12.5	14.3	15.7	17.2	18.6	19.8	21.1	22.2	23.3	24.4	25.4	26.4	27.3	28.2	29.1	29.9	30.8	31.6	32.4	33.9
42	12.7	14.5	16.1	17.6	19.0	20.3	21.6	22.7	23.8	24.9	25.9	26.9	27.9	28.8	29.8	30.7	31.4	32.2	33.0	34.5
44	13.0	14.8	16.4	18.0	19.4	20.7	22.0	23.1	24.3	25.4	26.5	27.5	28.5	29.5	30.3	31.2	32.1	32.9	33.7	35.3
46	13.3	15.1	16.7	18.4	19.8	21.1	22.4	23.6	24.8	25.9	27.0	28.1	29.1	30.1	31.0	31.9	32.8	33.8	34.6	36.2
48	13.5	15.4	17.0	18.7	20.1	21.5	22.8	24.1	25.2	26.4	27.5	28.6	29.6	30.5	31.6	32.5	33.4	34.3	35.2	37.0
50	13.7	15.7	17.3	19.0	20.4	21.9	23.2	24.5	25.7	26.9	28.0	29.2	30.3	31.3	32.2	33.1	34.1	35.0	35.9	37.6
52	13.9	15.9	17.6	19.2	20.8	22.2	23.6	24.9	26.2	27.4	28.5	29.6	30.7	31.8	32.9	33.8	34.7	35.6	36.5	38.3
54	14.1	16.1	17.9	19.6	21.1	22.6	24.0	25.3	26.6	27.8	29.0	30.1	31.2	32.3	33.4	34.4	35.3	36.3	37.2	38.9
56	14.3	16.3	18.2	19.9	21.5	22.9	24.4	25.7	27.0	28.3	29.5	30.6	31.7	32.8	33.9	34.9	35.9	36.9	37.8	39.6
58	14.6	16.6	18.4	20.2	21.8	23.3	24.7	26.1	27.4	28.7	30.0	31.1	32.2	33.3	34.4	35.4	36.4	37.4	38.4	40.3
60	14.7	16.8	18.7	20.4	22.1	23.6	25.1	26.5	27.8	29.1	30.5	31.6	32.7	33.8	34.9	36.1	37.1	38.1	39.1	40.9
62	15.0	17.0	19.0	20.7	22.4	24.0	25.5	26.9	28.2	29.5	30.9	32.1	33.2	34.3	35.4	36.6	37.7	38.7	39.6	41.6
64	15.1	17.3	19.2	21.0	22.7	24.3	25.9	27.3	28.6	29.9	31.3	32.6	33.7	34.8	35.9	37.1	38.2	39.2	40.2	42.2
66	15.3	17.5	19.5	21.2	23.0	24.6	26.2	27.7	29.0	30.3	31.7	33.0	34.2	35.3	36.4	37.6	38.7	39.8	40.8	42.8

*Additional sizes: $4 \times 5 = 4.9$; $4 \times 6 = 5.4$; $4 \times 7 = 5.8$; $5 \times 5 = 5.5$; $5 \times 6 = 6.3$; $5 \times 7 = 6.5$.

TABLE 1. CIRCULAR EQUIVALENTS OF RECTANGULAR DUCTS FOR EQUAL FRICTION—(Continued)

Side Rectangular Duct	26	28	30	32	34	36	38	40	42	44	46	48
26	28.6											
28	29.7	30.8										
30	30.7	31.9	33.0									
32	31.7	32.9	34.1	35.2								
34	32.7	33.9	35.1	36.3	37.4							
36	33.7	34.9	36.1	37.3	38.5	39.6						
38	34.6	35.9	37.1	38.4	39.5	40.7	41.8					
40	35.3	36.7	38.0	39.3	40.5	41.7	42.9	44.0				
42	36.0	37.6	39.0	40.3	41.5	42.7	44.0	45.1	46.2			
44	36.9	38.5	39.9	41.2	42.5	43.7	44.9	46.1	47.2	48.4		
46	37.8	39.3	40.8	42.2	43.5	44.8	46.0	47.2	48.4	49.5	50.6	
48	38.5	40.0	41.5	43.0	44.4	45.6	46.9	48.1	49.3	50.5	51.6	52.8
50	39.2	40.8	42.3	43.8	45.2	46.5	47.9	49.1	50.4	51.6	52.9	54.0
52	40.0	41.6	43.1	44.7	46.1	47.5	48.9	50.1	51.3	52.5	53.8	55.0
54	40.7	42.4	44.0	45.5	47.0	48.4	49.9	51.1	52.3	53.5	54.8	56.0
56	41.3	43.0	44.6	46.2	47.7	49.1	50.6	52.0	53.3	54.6	55.9	57.0
58	42.1	43.8	45.4	47.0	48.5	50.0	51.5	52.9	54.2	55.5	56.8	58.0
60	42.7	44.5	46.1	47.8	49.3	50.9	52.3	53.8	55.0	56.4	57.7	58.9
62	43.4	45.1	46.8	48.4	50.0	51.7	53.0	54.5	55.9	57.2	58.5	59.7
64	44.0	45.8	47.5	49.2	50.9	52.4	53.9	55.4	56.8	58.1	59.4	60.6
66	44.7	46.5	48.2	50.0	51.6	53.1	54.7	56.2	57.6	59.1	60.4	61.6
68	45.3	47.2	48.9	50.7	52.2	53.8	55.5	56.9	58.4	59.9	61.3	62.6
70	46.0	47.8	49.5	51.3	52.9	54.5	56.2	57.7	59.1	60.6	62.1	63.5
72	46.5	48.4	50.1	51.9	53.7	55.4	57.0	58.7	60.0	61.3	63.0	64.5

Side Rectangular Duct	50	54	60	66	72	78	84	88
50	55.0							
52	56.1							
54	57.2	59.4						
56	58.3	60.5						
58	59.3	61.6						
60	60.3	62.7	66.0					
62	61.3	63.7	67.1					
64	62.2	64.7	68.2					
66	63.2	65.7	69.3	72.6				
68	64.1	66.6	70.3	73.7				
70	65.0	67.6	71.3	74.8				
72	65.9	68.5	72.3	75.9	79.2			
74	66.8	69.4	73.3	76.9	80.3			
76	67.6	70.3	74.2	77.9	81.4			
78	68.4	71.2	75.2	78.9	82.5	85.8		
80	69.2	72.1	76.1	79.9	83.6	86.9		
82	70.1	73.0	77.1	80.9	84.6	88.0		
84	70.9	73.8	78.0	81.9	85.6	89.1	92.4	
86	71.7	74.6	78.9	82.9	86.6	90.2	93.5	
88	72.5	75.5	79.8	83.9	87.5	91.2	94.6	96.8
90	73.3	76.3	80.6	84.7	88.5	92.2	95.7	97.9
92	74.1	77.1	81.4	85.6	89.5	93.2	96.7	99.0
94	74.8	77.8	82.2	86.5	90.4	94.2	97.8	100.1
96	75.5	78.7	83.0	87.4	91.3	95.2	98.8	101.2

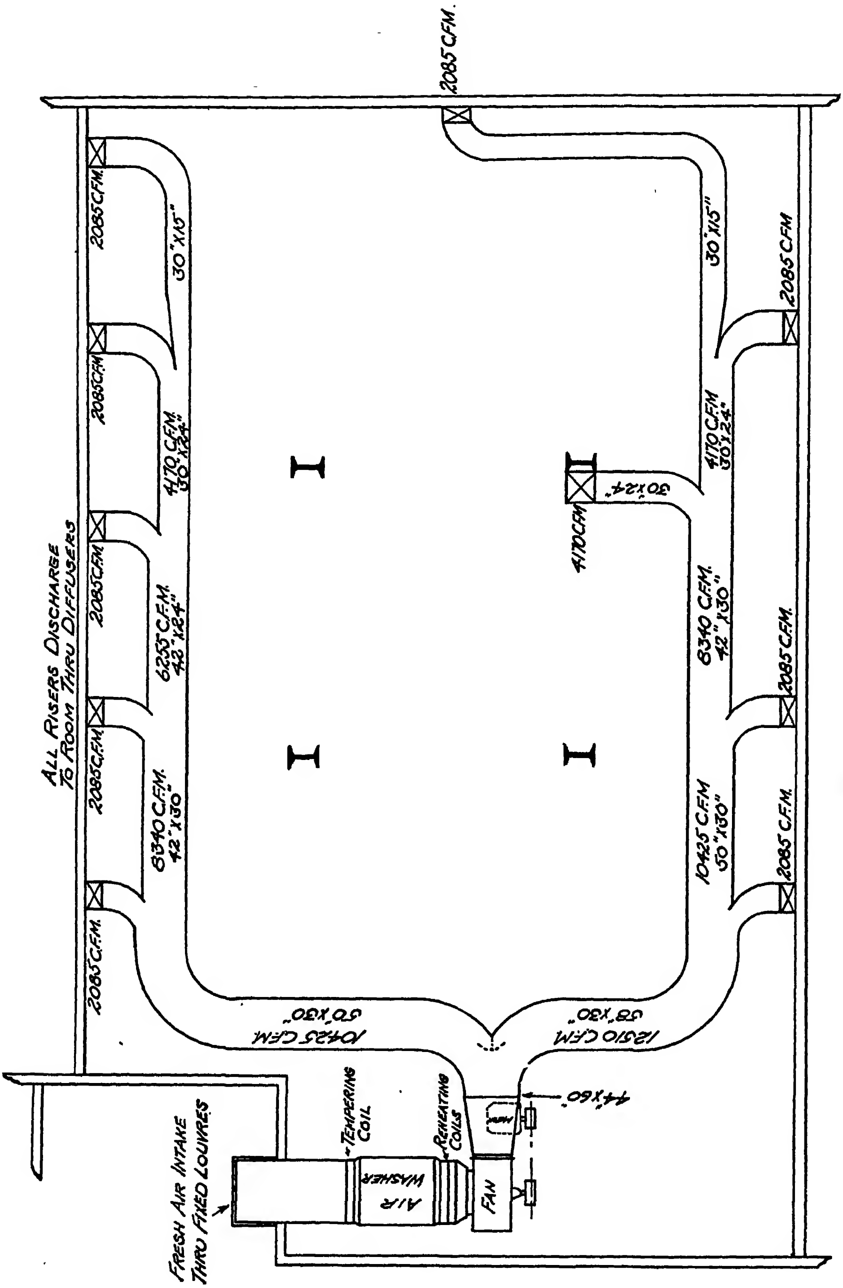

FIG. 6. TYPICAL LAYOUT OF AIR DISTRIBUTION SYSTEM

TABLE 2. PIPE SIZES FOR EXAMPLE 3

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
22,935	100.0	56	60 x 44
12,510	54.6	45	58 x 30
10,425	45.4	42	50 x 30
8,340	36.3	39	42 x 30
6,255	27.2	35	42 x 24
4,170	18.2	29½	30 x 24
2,085	9.1	23	30 x 15

[Velocity through diffusers (not shown) to be approximately 300 fpm].

The main pipe size should be selected to give a velocity equal to or less than the velocity at the fan outlet. Choosing a 56-in. pipe with a cross-sectional area of 17.1 sq ft, the velocity in the main pipe will be 1340 fpm. Using the friction pressure loss method this 56-in. main pipe will be taken as the basis of calculation.

Fig. 6 shows the amount of air to be handled by each section of pipe. Expressing the volume handled by each section as a percentage of the total volume and using the charts, Figs. 4 and 5, the pipe sizes are as shown in Table 2.

The pressure at the outlets nearest the fan will be greater than at the pipes farther along the run so that the former will tend to deliver more than the calculated amount of air. To remedy this condition, volume regulating dampers should be located at the base of each riser and adjusted for proper distribution. At points where branches leave the main it may be advisable, depending upon the nature of the installation, to install adjustable splitters similar to that shown in Fig. 6 where the main duct divides into the 58 in.-by-30 in. and 50 in.-by-30 in. branches.

The rectangular equivalents are selected from Table 1; the width to depth proportion will be determined by construction requirements and ease of fabrication. The calculation of the friction is as follows:

The longest run from the fan outlet to diffuser is 150 ft, 0 in.; 150 ft of 56-in. pipe is equivalent to $\frac{150 \times 12}{56}$ .... 32.2 dia.

Two 45-in., 90-deg elbows $(2 \times \frac{45}{56} \times 10)$ .... 16.1 dia.

Two 23-in., 90-deg elbows $(2 \times \frac{23}{56} \times 10)$ .... 8.2 dia.

Two 23-in., 90-deg elbows in riser $(2 \times \frac{23}{56} \times 30)$ .... 24.7 dia.

(Two bad elbows in riser, each equivalent to 30 diameters of duct).

Total diameter of 56-in. pipe 81.2

The velocity head corresponding to a velocity of 1340 fpm is $\left(\frac{1340}{4005}\right)^2 = 0.112$ in.

Taking 50 diameters as one head loss, then $\frac{81.2}{50} \times 0.112 = 0.182$ in. static loss in duct.

Where the connection pieces are made with long easy slopes and the general workmanship is good, a regain in static pressure may be deducted from the foregoing pressure loss. This can be taken as approximately two-thirds the difference in velocity pressures at the fan outlet and the last run of pipe. The velocity in the riser is 667 fpm with a corresponding velocity pressure of 0.033 in. The fan outlet velocity is 1400 fpm with a corresponding velocity pressure of 0.122 in. The regain equals two-thirds (0.122 − 0.033) = 0.059 in.

The net static pressure loss in the duct only is then:

0.182 in. − 0.059 in. .... 0.123 in.

Other friction losses are as follows:

(1) Fresh air intake 1000-fpm velocity (1½ heads × 0.0625) .... 0.094 in.
(2) Tempering coil loss (from manufacturers tables) .... 0.100 in.
(3) Air washer loss (from manufacturers tables) .... 0.250 in.
(4) Reheating coil loss (from manufacturers tables) .... 0.100 in.
(5) Allowance for regulating dampers and diffusers .... 0.100 in.

Static pressure loss of system .... 0.767 in.

The fan should be selected from the manufacturers ratings which, according to the Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers[1], will deliver 22,935 cfm at a static pressure of 0.767 in. and which has an outlet area of 16½ sq ft.

The method of design used in Example 3 is the *equal friction method* described under the heading Procedure for Duct Design. This involves the arbitrary reduction of velocity from the fan outlet to the point of discharge to the room, and the friction is calculated by adding the pressure losses of each section of duct. This method requires dampering in the risers.

Example 4. Fig. 7 shows an exhaust system layout for exhausting from buildings of the same type as in Example 3. Assume the air requirements based on the number of air changes per hour to be 16,800 cfm. Using a velocity of 1400 fpm in the main duct at the fan inlet, which is an average velocity for this type of system, the area of the main is 12 sq ft, which corresponds to a 47-in. pipe. Referring to Example 3, and using the charts, Figs. 4 and 5, the pipe sizes are as indicated in Table 3.

All risers will require dampering as in Example 3. The calculation of the friction is as follows:

The longest run from the intake grille to fan inlet is 100 ft.

(1) Duct friction 100 ft of 47-in. pipe $\left(\frac{100 \times 12}{47}\right)$ .... 25.6 dia.

Two 28½-in., 90-deg elbows in riser $\left(\frac{2 \times 28.5 \times 30}{47}\right)$ .... 36.4 dia.

(Two bad elbows in riser each equivalent to 30 diameters of duct).

TABLE 3. PIPE SIZES FOR EXAMPLE 4

VOLUME OF AIR (CFM)	PER CENT OF TOTAL VOLUME	DIAMETER OF PIPE (INCHES)	EQUIVALENT SIZE OF RECTANGULAR DUCT (INCHES)
16,800	100.0	47	38 x 48
11,550	68.8	41	30 x 46
9,450	56.2	38	30 x 40
5,250	31.3	31	24 x 34
4,200	25.0	28.5	24 x 28
3,150	18.8	25.3	16 x 34
2,100	12.5	21.6	16 x 24

[Velocity through intake grilles (not shown) to be approximately 400 fpm].

[1] See Chapters 17 and 42.

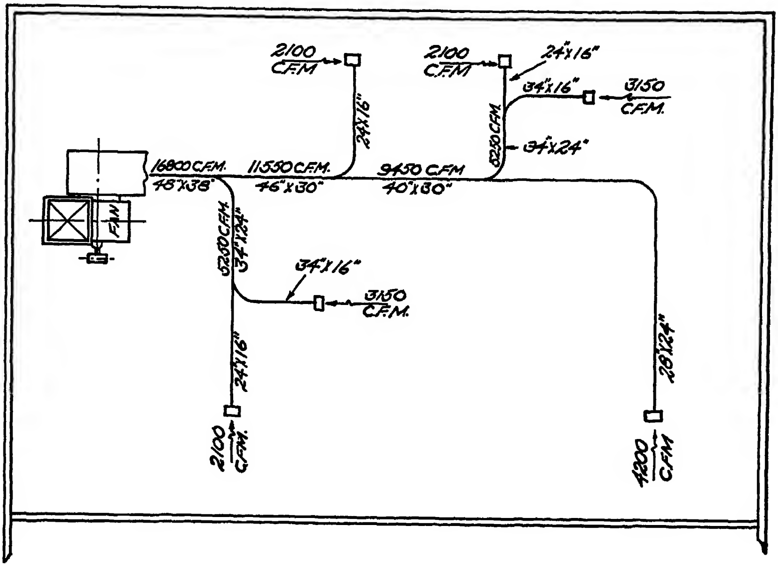

FIG. 7. EXHAUST SYSTEM LAYOUT

One 28½-in., 90-deg elbow in horizontal run $\left(\frac{28.5 \times 12}{47}\right)$	6.0 dia.
Total diameter of 47-in. pipe	68.0 dia.

Velocity head corresponding to 1400 fpm is $\left(\frac{1400}{4005}\right)^2 = 122$ in.

Taking 50 diameters as one head loss, then $\frac{68 \times 0.122}{50}$	0.166 in.
(2) Intake loss from grille (1½ heads at a 400 fpm velocity $1\frac{1}{2} \times 0.01$)	0.015 in.
(3) Static pressure required to produce one velocity head at 1400 fpm	0.122 in.
(4) Loss occasioned by step-up of velocity (0.20×0.122)	0.024 in.
(This loss varies from 0.05 to 0.40 velocity heads depending upon the nature of the change. For average systems 0.20 velocity heads is a close approximation).	
Static pressure loss on inlet side	0.327 in.

To this must be added the resistance on the discharge side of the fan. A fan outlet velocity of approximately 1500 to 1600 fpm may be used. Assuming the fan outlet to be equivalent in area to a 45-in. pipe, the velocity is 1525 fpm.

Loss on discharge (15 ft from fan outlet to discharge):

$$\frac{15 \times 12}{45} = 4 \text{ diameters of 45-in. pipe.}$$

The velocity head corresponding to a velocity of 1525 fpm is 0.145 and the discharge-side loss is $\frac{0.145 \times 4}{50} = 0.012$ in. The total static pressure loss of the system is then:

$$0.012 + 0.327 = 0.339 \text{ in.}$$

The fan will be selected to handle 16,800 cfm at a static pressure of 0.339 in. and to have an outlet velocity of 1525 fpm. Outlet area 11 sq ft.

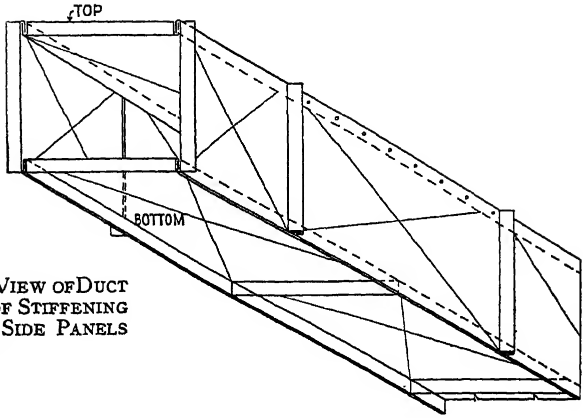

Fig. 8. Isometric View of Duct Showing Location of Stiffening Seams on Top and Side Panels of Duct

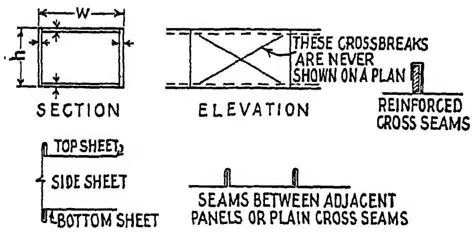

Fig. 9. Details of Seams

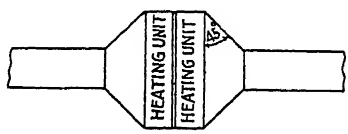

Fig. 10. Method of Installing Heating Unit

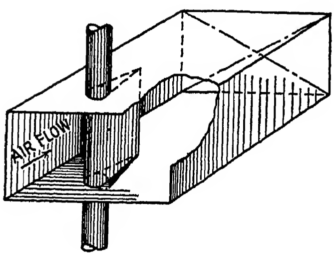

Fig. 11. Installation of Easement in Duct Around Obstruction

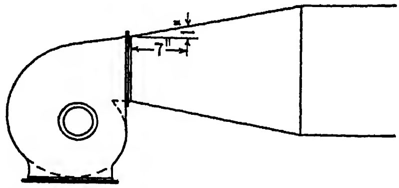

Fig. 12. Fan Discharge Connection

Where there are one or more ducts with branches, the velocity of air in the ducts may be either chosen arbitrarily or calculated for friction losses. When arbitrary values are assigned, a certain amount of dampering should be provided for; this will be small when the method chosen permits a drop in velocity as the quantity of air is reduced.

After the total air quantity and the size of fan are ascertained, the main duct is usually fixed as being at least equal in area to the fan outlet, or perhaps 10 per cent greater. From this main pipe all others are proportioned. For example, if the main duct is 30 in. in diameter, a branch to carry 10 per cent of the total capacity should be 12.7 in. in diameter (see Fig. 4) in order to have the same friction per foot of length, while one carrying one-half the total capacity of a 30-in. main with the same friction loss per foot would be 23.4 in. in diameter. By this method of equalizing friction it is unnecessary to consider the resistance of each section of pipe independently, but only to know the distance from the fan outlet to the end of the longest run of pipe, the number and size of elbows, and the diameter and velocity in the largest pipe.

Example 5. If the greatest length of piping in a system is 130 ft with a 26 in. diameter main pipe and one 20-in. elbow, the piping having been designed for equal friction per foot of length, the friction would be the same as for 130 linear feet of 26 in. pipe, or 60 diameters. To this should be added the friction loss in elbows, in this case one 20 in. elbow, which has a loss equivalent to one-fifth of a velocity head or ten diameters of 20-in. pipe. This in turn is $\frac{20}{26} \times 10 = 7.7$ diameters of 26 in. pipe. The total equivalent length of the system will then be $60 + 7.7$, or 67.7 diameters. Since 50 diameters is equivalent to one velocity head, the loss is $\frac{67.7}{50} = 1.35$ times the velocity head. If the velocity is, for example, 2200 fpm, corresponding to 0.3 in. pressure, the friction loss of the system will be $1.35 \times 0.3 = 0.405$ in.

Frequently the prevention of sound in a heating or ventilating system imposes more severe restrictions than the prevention of excessive pressure drop. This question is highly involved and requires consideration of many factors. The air velocities to be used will vary with the standard of construction used in the ducts themselves as well as with the nature of the occupancy and the construction of the building. In general, architects and engineers who leave the details of duct construction to the contractor must, of necessity, design for lower velocities than might be required for quiet operation if proper construction details were always followed. The contractor may be expected to build the ducts by the least expensive methods, and the engineer must anticipate this. For further information on noise reduction, see Chapter 18.

Details of Duct Construction

If panel construction is used with standing seams or similar reinforcement, and the panels are cross-broken to give rigidity, there is less likelihood of vibration due to air flow, or deflection due to air pressure. Elbows made without splitters, and improperly shaped transformation sections produce high local velocities which are the cause of noise in duct work. The use of first-class duct construction with well designed transformation sections and splitters in elbows tends to maintain relatively uniform velocities with decrease in turbulence and in the noise produced.

Figs. 8 to 15 show acceptable construction details for rectangular ducts, elbows, transformation pieces or connections, and air splitters. Other methods are also acceptable, such as the use of angle iron stiffeners for large ducts. Good construction is essential to the elimination of duct noises and for the prevention of a flimsy installation.

Fig. 8 is an isometric view of a duct showing the location of the stiffening seams on the top and side panels. The cross seams should not

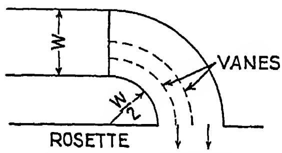

FIG. 13. AIR SPLITTERS INSTALLED IN ELBOW

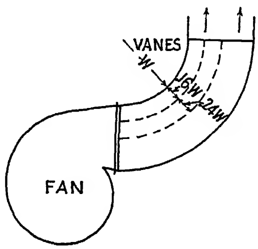

FIG. 14. AIR SPLITTERS INSTALLED IN ELBOW AT FAN DISCHARGE

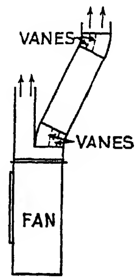

FIG. 15. AIR SPLITTERS IN BRANCH DUCTS AND ELBOWS

occur at the same place but should be staggered as indicated. Heating units should be installed as shown in Fig. 10 with the duct connections making an angle of not less than 45 deg, but preferably 60 deg. Fan discharge connections should have a maximum slope of 1 in 7, as indicated in Fig. 12. Whenever a pipe or other obstruction passes through a duct an easement should be placed around the pipe as indicated in Fig. 11. Air splitters should be installed in elbows as shown in Figs. 13 and 14. The recommended gages for rectangular sheet metal duct construction are given in Table 4.

TABLE 4. SHEET METAL GAGES FOR RECTANGULAR DUCT CONSTRUCTION[a]

GAGE	WIDTH OF DUCT	SEAM	REINFORCED SEAM
26	Up to 12 in.		
24	13 in. to 30 in.	1	
22	31 in. to 48 in.	1	
22	49 in. to 60 in.	1½	⅛ in. x 1⅜ in.
20	61 in. to 90 in.	1½	⅛ in. x 1⅜ in.

[a]If panels are not cross-broken two gages heavier material should be used.

Chapter 20

AIR DISTRIBUTION

Warm Air Systems, Combined Systems, Split Systems, School Buildings, Theaters, Upward System, Downward System, Vanes

TO produce proper air distribution in a room to be ventilated, heated, or cooled by air, the design and location of the air supply inlets and exhaust outlets must be carefully considered. Systems fail though they handle the proper amount of air, because important design principles are ignored.

WARM AIR SYSTEMS

With gravity warm air systems, it has been the practice to place the supply registers in or near the floor of each room and to place the return grille in the floor of the first story. When there is mechanical air circulation, the supply ducts are extended to the outside walls and the air is discharged into the rooms near their cold exposures; on the return side a grille is placed in or near the floor at a central location, or individual return grilles are provided, usually at the side of the room opposite the supply register.

These arrangements are usually satisfactory for heating (Fig. 1) but not for cooling (Fig. 2). If cool air is introduced at one side of the room at the floor, and if the escape opening for the heated air to be displaced by the cool air is at the floor at the other side, the cool air will travel across the floor and will escape through the vent or return air opening, and thus not appreciably affect the over-heated air in the upper part of the room.

The air supply opening will serve satisfactorily if located high on an interior wall opposite the exposed wall, and this location answers well also for gravity indirect heating. The corresponding return air arrangements, however, apparently are not subject to exact rules, but must be adapted to circumstances. For example, where the building is compact, with a first story having rooms open to each other, a single, centrally-located return at the floor functions satisfactorily for heating, and if the second story bedrooms are also compactly arranged no individual return from each will be necessary. On the other hand, any room which is unusually exposed, which is especially remote with reference to the other rooms, or which is apt to be tightly closed most of the time, should have a controlled return grille and duct. With a mechanical warm air system, this return may be close to the floor below the supply grille, and with a gravity system may be close to the floor at the opposite side of the room from the supply grille.

COMBINED SYSTEMS

For a combined mechanical heating and cooling system using refrigeration for cooling, no particular change in the ducts usually is necessary. It is desirable from an economic standpoint to take advantage of the natural tendency of the cooler air to remain below the warmer air overhead, and anything which will bring about such stratification will effect an economy in refrigeration.

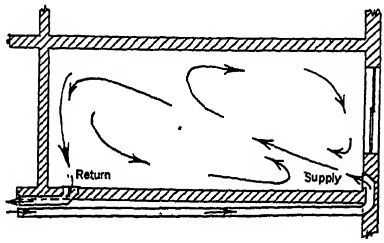

FIG. 1. AIR CIRCULATION WHEN HEATING WITH LOW-SUPPLY AND RETURN OPENINGS

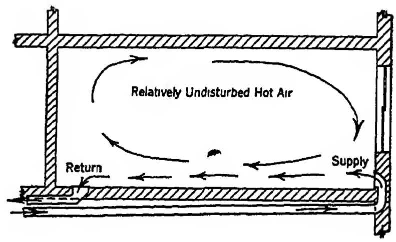

FIG. 2. AIR CIRCULATION WHEN COOLING WITH LOW-SUPPLY AND RETURN OPENINGS

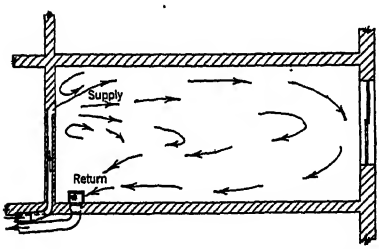

FIG. 3. AIR CIRCULATION WHEN COOLING WITH HIGH-SUPPLY OPENING AND LOW-RETURN OPENINGS

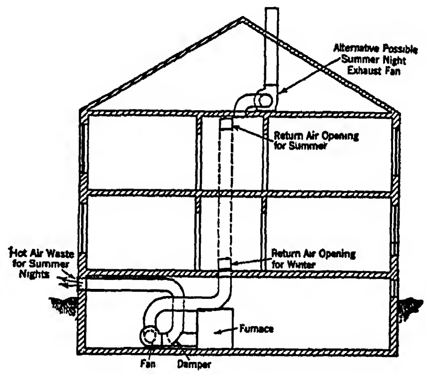

FIG. 4. SECTION THROUGH AN ELEMENTAL MECHANICAL WARM AIR HEATING-COOLING SYSTEM. THE ATTIC FAN IS ALTERNATIVE

If the return ducts of a mechanically operated warm air system are adequate, appreciable cooling can be accomplished with natural means, as follows:

The fan outlet usually has a by-pass duct leading to a basement window or to a chimney provided for the purpose. The return duct has an alternative shaft opening into the highest part of the house. At night, in summer, the fan may be operated to exhaust the hot air from the top of the house by the return air duct just described and the fan will blow this heated air out of doors through the window, or preferably, of course, through the chimney. The cooler night air must then enter the house through the windows, and by its motion and temperature will extract the heat from the walls and furniture.

Fig. 3 shows the air circulation when cooling with a high supply opening and a low return opening. The air circulation, when heating, will be substantially the same as when cooling. Fig. 4 shows a section through an elemental mechanical warm air heating-cooling system. The attic plan is alternative. Summer night cooling may, of course, be accomplished by placing an exhaust fan in the attic.

SPLIT SYSTEMS

Many buildings which are heated by radiators or convectors and which have rooms requiring ventilation or cooling have air supply and exhaust systems independent of the radiators or convectors. Such installations are termed *split systems*. When the air enters a room through conventional side wall inlets an occupant may feel comfortable if the air is about the temperature of the room, but the introduction of too cool air may cause a feeling of draft. To correct this draft condition, glass chutes and elaborate diffusers are sometimes provided. The arrangement shown in Fig. 5 for supplying cool air to a room supplies satisfactory air circulation in spaces up to 400 sq ft in area with ceilings as low as 8 ft. There is no maximum ceiling limitation as to height.

When the room in question is provided with a unit ventilator which obtains its air supply directly through the wall from out of doors, the problem of distribution is by no means easy, although with a high velocity air jet passing in an upward direction, satisfactory air distribution will be had.

The use of unit air conditioners for summer cooling introduces no new features or difficulties which have not already been encountered in winter heating. Conditioners must be provided with positive control by means of valves or dampers, or both, which will prohibit any sudden and wide temperature variation, and keep the entering air not more than approximately 7 deg cooler than the air already in the space. This temperature margin is dependent on various factors including the ceiling height of the room and the velocity of the air at the discharge grille.

SCHOOL BUILDINGS

The air distribution conditions in school building classrooms are not unlike those illustrated in Fig. 1 for mechanical warm air systems and those in Fig. 6 for unit ventilator-equipped plants. The thermostat (Fig. 6) which controls the mixing damper and the heating unit in the unit ventilator should be in the air stream from the machine. School rooms which have center-ceiling inlets along the lines of Fig. 5 have given excellent results. It is important that the temperature of the entering air, whether this air be supplied by a local unit ventilator or by a distant central fan, be controlled so that the air cannot enter the room from a side-wall inlet or from a unit ventilator at a temperature more than a very few degrees cooler than that of the air already near the ceiling of the room.

Fig. 7 shows a section through a room equipped with a unit air conditioner or unit cooler. This is typical of the condition in effect when any recirculating room-cooling unit is installed.

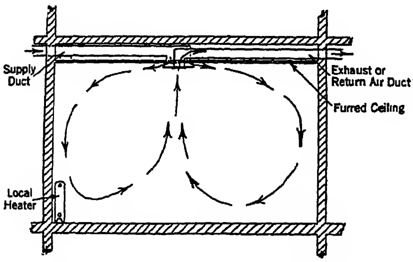

FIG. 5. SECTION THROUGH A RADIATOR-HEATED ROOM

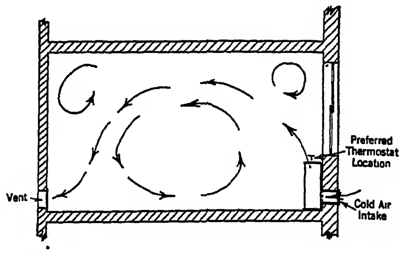

FIG. 6. SECTION THROUGH A UNIT VENTILATOR-EQUIPPED ROOM WHEN HEATING

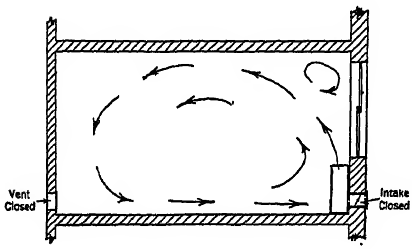

FIG. 7. SECTION THROUGH A UNIT CONDITIONER-EQUIPPED ROOM WHEN COOLING

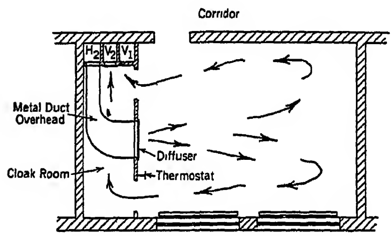

FIG. 8. PLAN OF A CLASSROOM IN A SCHOOL VENTILATED BY A CENTRAL FAN

In Fig. 8 the cloakroom ceiling is furred down so as to conceal the metal air supply duct, which is close to the ceiling. The air for ventilation usually is controlled by a duct thermostat near the fan, at a temperature slightly higher than the temperature required in the room, to allow for heat losses in the duct system.

THEATERS

Theaters are usually ventilated or cooled by introducing pre-conditioned air. No ventilating system for a theater should be given consideration without definite provision for cooling. Theater cooling generally is far more important than theater heating. There are two widely different methods of theater air distribution, the *upward* and the *downward*.

Upward System

If no inlet openings are possible in the ceiling, the upward system may be the least objectionable alternative. Fig. 9 shows a section through a theater with the upward system of air distribution. The occupants often suffer from drafts due to the cool air which comes from the unoccupied zones.

When the entire seating area is occupied, the upward system gives little trouble when cooling, and since very little heating is required under such conditions, practically no difficulty is encountered. The maximum

volume of air to be introduced with the upward system is about 25 cfm of air per person at a low velocity, say at 150 fpm (linear), and at a temperature not more than 6 deg below the room temperature. For partial occupancy, higher entering air temperatures can be used, with correspondingly less danger from drafts.

Downward System

Theaters usually are equipped with downward air distribution with horizontal diffusion of the entering cool air so as to combine it, both as to temperature and dilution, with the heated air which inevitably must rise from the bodies of the patrons. The waste or the recirculated air is withdrawn from the room at the floor. If the theater is large, and if the exhaust openings are placed in the side walls at the floor, drafts may be felt by the people who sit near the openings. There is no objection, however, except that of cost, to the use of small exhaust openings under each seat. These may be cleanable floor grilles or may have mushroom covers.

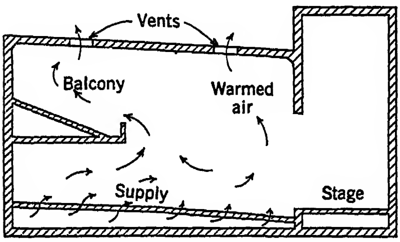

FIG. 9. THEATER WITH UPWARD SYSTEM OF VENTILATION

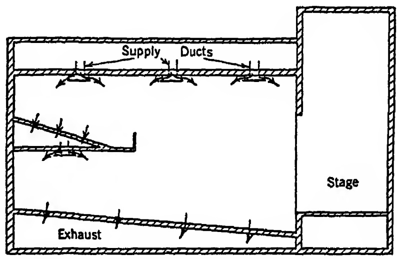

FIG. 10. SECTION THROUGH A THEATER WITH DOWNWARD VENTILATION

In a downward system, if the entering cool air is not deflected horizontally, it will fall through the surrounding much hotter air, and will reach high velocities by the time it strikes the heads of the occupants. Air at a temperature 10 deg below that of the surrounding air is decidedly objectionable when forced over one's head at a velocity of nearly 400 fpm.

Fig. 10 shows a section through a theater with downward ventilation. The deflectors cause the entering cool air to be spread horizontally so that it will mix with the hotter air. The final escape is through well-distributed openings in the floor. There have been cases in which the downward system of air distribution such as that illustrated in Fig. 10 gave trouble due to overheating at the rear, both above and below the balcony, especially when not provided with refrigeration for cooling, and when not adequately controlled. It is especially necessary that adequate removal of the heated air be provided at these low-ceiling points and it is probable that auxiliary exhaust at or through the ceiling after the manner of the arrangements shown in Fig. 5 would be helpful.

VANES

In order to cause the supply air to a room to take a fixed or desired direction when leaving the inlet opening of a flue, stationary vanes may

be provided at both the back of the grille and at the grille to direct the air flow. Fig. 11 shows a section through a room inlet opening at the top of a rising flue and indicates the air conditions when no vanes are used. Fig. 12 shows a section through the same room inlet opening when vanes are advantageously placed to direct the flow of air.

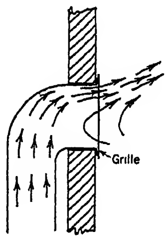

FIG. 11. AIR CONDITIONS AT INLET OPENING AT THE TOP OF A RISING FLUE WHEN NO VANES ARE USED

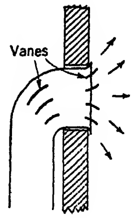

FIG. 12. AIR CONDITIONS AT INLET OPENING AT THE TOP OF A RISING FLUE WHEN DIRECTIONAL VANES ARE USED

In many theater and commercial installations the ejector-like action of high-velocity air emerging from a duct is taken advantage of, and scientifically proportioned nozzles are installed to cause definite recirculation of the room air.

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Chapter 21

INDUSTRIAL EXHAUST SYSTEMS

Types, Design of Systems, Suction and Velocity Requirements, Design of Hoods, Design of Duct Systems, Collectors, Resistance of Systems, Selection of Fans and Motors

EXHAUST and collecting systems are found in almost every industry and are a vital adjunct in maintaining safe and hygienic conditions. The present chapter attempts to give general information relating to the design of factory exhaust systems in order that efficient and economical control of dusts and fumes may be achieved.

TYPES OF SYSTEMS

There are two general arrangements, the central and the group systems. In the central system a single or double fan is located near the center of the shop with a piping system radiating to the various machines to be served. In the group system, which is sometimes employed where the machines to be served are widely scattered, small individual exhaust fans are located at the center of the machine groups. The group arrangement has the advantage of flexibility.

Exhaust systems are also classified by the means employed to collect dust or other material handled. The dust or refuse may be collected and controlled by enclosing hoods, open hoods, inward air leakage, or by exhausting the general air of the room.

With some classes of machinery it is not feasible to closely hood the machines and in these cases open hoods over or adjacent to the machines are provided to collect as much as possible of the dust and fumes. This class includes such machines as rubber mills, package filling machinery, sand blast, crushers, forges, pickling tanks, melting furnaces, and the unloading points of various types of conveyors.

The open hoods should be placed as close to the source of dust or fumes as possible, with due regard to the movements of the operator. When the hood must be placed at some distance above the machine it should be large enough to encompass an area of considerable extent as diffusion is usually quite rapid.

Consideration must also be given to the natural movement of the fumes. For those that are lighter than air the hood should be over or above the machine and where a heavy vapor or dust-laden air at ordinary temperature is to be removed, horizontal or floor connections are required. If it is attempted to remove heavy dust such as lead oxides by an overhead hood the conditions may be worse than if no exhaust were used at

all, owing to the rising air current carrying the dust up through the breathing zones. The objective to keep in mind in all cases is to take advantage of the natural tendency of the material to move upward or downward.

In another class of operation the main objective is to prevent the escape of dust into the surrounding atmosphere, the removal of some dust from the machine or enclosure being merely incidental. The dust-creating apparatus is enclosed within a housing which is made as tight as practicable, and sufficient suction is applied to the enclosure to maintain an inward air leakage, thus preventing escape of the dust. While the exhaust system is only required to handle the air which leaks in through the crevices and openings in the enclosure, yet in many installations leakages are very high and great care is required to obtain satisfactory results with a system of this kind. The inward-leakage principle is utilized for controlling dust in the operating of tumbling barrels, grinding, screening, elevating and similar processes.

Certain dust and fume producing operations are best carried on by isolating the process in a separate compartment or room and then applying general ventilation to this space. The compartment or room in which the work is performed should be as small as is consistent with convenience in handling the work. The ventilating system should be designed so that a strong current of clean air is drawn across the operator, and away from him toward the work, where the dust is picked up and carried from the room.

DESIGN OF SYSTEMS

The first step in the design of an exhaust system is to determine the number and size of the hoods and their connections. No general rules, however, can be given since hood and duct dimensions are determined by the characteristics of the operations to which they are applied. When a tentative decision regarding the set-up has been made, it is then necessary to obtain the suction and air velocities required to effect control. At this point the designer must rely upon the prevailing practice and on such physical data relating to hoods, duct systems and collectors as are available. Finally, in choosing the fan, the area of the intake should be equal to or greater than the sum of the areas of the branch ducts. The speed, of course, must be sufficient to maintain the estimated suction and air velocities in the system. In general, the most important requirements of an efficient exhaust and collecting system are as follows[1]:

1. Hoods, ducts, fans and collectors should be of adequate size.

2. The air velocities should be sufficient to control and convey the materials collected.

3. The hoods and ducts should not interfere with the operation of a machine or any working part.

4. The system should do the required work with a minimum power consumption.

5. When inflammable dusts and fumes are conveyed, the piping should be provided with an automatic damper in passing through a fire-wall.

[1]For more detailed requirements see Safe Practice Pamphlets Nos. 32 and 37, published by the *National Safety Council*, Chicago.

6. Ducts and all metal parts should be grounded to reduce the danger of dust explosions by static electricity.

7. The design of an exhaust system should afford easy access to parts for inspection and care.

SUCTION AND VELOCITY REQUIREMENTS

The removal of dust or waste by means of an exhaust hood requires a movement of air at the point of origin sufficient to carry them to a collecting system. The air velocities necessary to accomplish this depend upon the physical properties of the material to be eliminated and the

TABLE 1. SIZE OF CONNECTIONS FOR WOOD-WORKING MACHINERY

TYPE OF MACHINE	DIAMETER OF CONNECTIONS IN INCHES
Circular Saws, 12-in. diam.	4
Circular Saws, 12-24-in. diam.	5
Circular Saws, 24-40-in. diam.	6
Band Saws, Blade under 2 in. wide	4
Band Saws, Blade 2-3 in. wide	5
Band Saws, Blade 3-4 in. wide	6
Band Saws, Blade 4-5 in. wide	7
Band Saws, Blade 5-6 in. wide	8
Small Mortisers	6
Single End Tenoners	6
Double End Tenoners	7
Double End, Double Head Tenoners	10
Planers, Matchers, Moulders, Stickers, Jointers, etc.—	
With Knives, 6-10 in.	5–6
With Knives, 10-20 in.	6–8
With Knives, 20-30 in.	6–10
Shapers, Light Work	4–5
Shapers, Heavy Work	8
Belt Sander, Belt less than 6 in. wide	5
Belt Sander, Belt 6-10 in. wide	6
Belt Sander, Belt 10-14 in. wide	7
Drum Sander, 24 in.	5
Drum Sander, 30 in.	6
Drum Sander, 36 in.	7
Drum Sander, 48 in.	8
Drum Sander, over 48 in.	10
Disc Sander, 24 in. diam.	5
Disc Sander, 26-36 in. diam.	6
Disc Sander, 36-48 in. diam.	7
Arm Sander	4

direction and speed with which it is thrown off. If the dust to be removed is already in motion, as is the case with high-speed grinding wheels, the hood should be installed in the path of the particles so that a minimum air volume may be used effectively. It is always desirable to design and locate a hood so that the volume of air necessary to produce results is as small as possible.

The static suction at the throat of a hood is frequently used in practice as a measure of the effectiveness of control. This is of considerable value where exhaust systems adapted to particular operations have been standardized by practice. Tables 1 and 2 present the duct sizes usually

employed for standard wood-working machinery and for grinding and buffing wheels. Static pressures which in practice have been found necessary to control and convey various materials, are given in Table 3. It must be remembered, however, that the term *suction* is merely a rough

TABLE 2. SIZE OF CONNECTIONS FOR GRINDING AND BUFFING WHEELS

DIAMETER OF WHEELS	MAX. GRINDING SURFACE SQ IN	MIN. DIAM. OF BRANCH PIPES IN INCHES
Grinding—		
6 in. or less, not over 1 in. thick	19	3
7 in. to 9 in., inclusive, not over 1½ in. thick	43	3½
10 in. to 16 in., " " " 2 in. "	101	4
17 in. to 19 in., " " " 3 in. "	180	4½
20 in. to 24 in., " " " 4 in. "	302	5
25 in. to 30 in., " " " 5 in. "	472	6
Buffing—		
6 in. or less, not over 1 in. thick	19	3½
7 in. to 12 in., inclusive, not over 1½ in. thick	57	4
13 in. to 16 in., " " " 2 in. "	101	4½
17 in. to 20 in., " " " 3 in. "	189	5
21 in. to 27 in., " " " 4 in. "	338	6
27 in. to 33 in., " " " 5 in. "	518	7

TABLE 3. SUCTION PRESSURES REQUIRED AT HOODS

	STATIC SUCTION IN INCHES OF WATER
Exhausting from grinding and buffing wheels	1½–5
Exhausting from tumbling barrels	2
Exhausting from wood-working machinery—light duty	2
Exhausting from wood-working machinery—heavy duty	2–4
Shoe machinery exhaust	2–3
Exhausting from rubber manufacturing processes	2
Flint grinding exhaust	2
Exhausting from pottery processes	2
Lead dust and fume exhaust	2–4
Fur and felt machinery exhaust	2–3
Exhausting from textile machinery	2–3
Exhausting from elevating and crushing machinery	2
Conveying bulky and heavy materials	3–5

measure of the air volume handled and consequently of the air velocity at the opening of the hood. The elimination of any dusty condition requires added information concerning the shape, size and location of the hood used with regard to the operation in question.

In some states grinding, polishing and buffing wheels are subject to regulation by codes. The static suction requirements, which range from 1½ to 5 in. water displacement in a *U*-tube, should be followed although in several instances they may appear to be excessive. Frequently, in these operations, a large part of the wheel must be exposed and the dust-

laden air within the hood is thrown outward by the centrifugal action of the wheel, thus counteracting useful inward draft. This tendency may be diminished by locating the connecting duct so as to create an air flow of not less than 200 fpm about the lower rim of the wheel.

Exact determinations of hood control velocities are not available, but it is safe to assume that for most dusty operations they should not be less than 200 fpm at the point of origin. For granite dust generated by pneumatic devices, Hatch[2] gives velocities from 150 to 200 fpm, depending on the type of hood used, as sufficient for safe control. Considering the character of the industry, air velocities of this order may be extended to similar dusty operations. The method for approximately determining these velocities in terms of the velocity at the hood opening is given below.

DESIGN OF HOODS

No set rule can be given regarding the shape of a hood for a particular operation, but it is well to remember that its essential function is to create an adequate velocity distribution. The fact that the zone of greatest effectiveness does not extend laterally from the edges of the opening may frequently be utilized in estimating the size of hood required. Where complete enclosure of a dusty operation is contemplated, it is desirable to leave enough free space to equal the area of the connecting duct. Hoods for grinding, polishing and buffing should fit closely, but at the same time should provide an easy means for changing the wheels. It is advisable to design these hoods with a removable hopper at the base to capture the heavy dusts and articles dropped by the operator. Such provisions are of assistance in keeping the ducts clear. Air volumes used to control many dust discharges may often be reduced by effective baffling or partial enclosure of an operation. This procedure is strongly urged where dusts are directed beyond the zone of influence of the hood.

Axial Velocity Formula for Hoods

When the normal flow of air into a hood is unobstructed, the following formula may be used to determine the air velocity at any point along the axis:

$$\frac{Y}{100 - Y} = \frac{0.1A}{x^2} \tag{1}$$

where

Y = per cent of velocity at opening.
A = area of opening in square inches (or square feet).
x = distance outward from opening in inches (or feet).

It is important to note that the velocity function varies in direct proportion to the area. Hence, under certain conditions, a large opening may function more effectively than a small one for the same volume of

[2]Control of the Silicosis Hazard in the Hard Rock Industries. (*Journal of Industrial Hygiene*, Vol. XII, No. 3, March, 1930).

flow. The formula, of course, presumes that the air velocity distribution across the hood opening is uniform[3].

Example 1. A small hood 64 sq in. in area handles 400 cfm. What will be the air velocity at a point 5 in. outward along the axis if the flow is unobstructed?

Solution. Substitute in Equation 1 and solve for Y, thus

$$\frac{Y}{100 - Y} = \frac{0.1 \times 64}{5 \times 5}$$

from which $Y = 20.4$ per cent of the velocity at the opening of the hood.

$$\text{Velocity at opening} = \frac{400 \times 144}{64} = 900 \text{ fpm}$$

Hence, the velocity at the point in question is $900 \times 0.204 = 184$ fpm

Air Flow from Static Readings

The volume of air flow into any hood may be determined from the following equation:

$$Q = 4005\, fa \sqrt{h_t} \tag{2}$$

where

Q = volume of air flow in cubic feet per minute.

a = area of connecting duct in square feet.

h_t = static suction at throat of hood in inches of water.

f = orifice or restriction coefficient which varies from 0.6 to 0.9 depending on the shape of the hood.

An average value of f is 0.71, although for a well-shaped opening a value of 0.8 may be used. If it is assumed that the entrance loss of a hood is proportional to the velocity head, f can be determined by the relation:

$$f = \sqrt{\frac{h_v}{h_v + h_t}} \tag{3}$$

where

h_v = the velocity head.

For duct ends and abrupt openings $h_t = h_v$ and for flared openings $h_t = 0.5h_v$.

The term *static suction* is not a good measure of the effectiveness of a hood unless the area of the opening and the location of the operation with respect to the hood are known. This is clearly indicated by Equation 1 which shows that the velocity function at any point along the axis varies directly as the area of the opening and inversely as the square of the distance. However, this formula coupled with Equation 2 should serve to indicate the velocity conditions to be expected when operations are conducted external to the hood opening.

Large Open Hoods

Large hoods, such as used for electroplating and pickling tanks, should be subdivided so that the area of the connecting duct is not less than one-

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[3]Velocity Characteristics of Hoods under Suction, by J. M. Dallavalle (A.S.H.V.E. TRANSACTIONS Vol. 38, 1932).

fifteenth the open area of the hood. Frequently, it will be found necessary to branch the main duct in order to obtain a uniform distribution of flow. *Canopy hoods* should extend 6 in. laterally from the tank for every 12-in. elevation. In most cases, hoods of this type take advantage of the natural tendency of the vapors to rise, and air velocities may be kept low. Cross drafts from open doors or windows disturb the rise of the vapors and therefore provision must be made for them. The air velocities required also depend upon the character of the vapors given off, cyanide fumes, for example, requiring an air velocity of approximately 75 fpm on the surface of the tank and acid and steam vapors requiring velocities as low as 25 to 50 fpm. The total volume of air flow necessary to obtain these velocities may be approximately determined from the following simple formula:

$$Q = 1.4PDV \tag{4}$$

where

Q = total volume of air handled by hood in cfm.

P = perimeter of the tank in feet.

D = distance between tank and hood opening in feet.

V = air velocity desired along edges and surface of tank in fpm.

Spray Booths

In the design of an efficient spray booth, it is essential to maintain an even distribution of air flow through the opening and about the object being sprayed. While in many instances, spraying operations can be performed mechanically in wholly enclosed booths, the volatile vapors may reach injurious or explosive concentrations. At all times, the concentrations of these vapors, and particularly those containing benzene, should be kept below 100 ppm. Spray booth vapors are dangerous to the health of the worker and care should be taken to minimize exposure to them.

It is recommended in the design of spray booths that the exhaust duct be located in a horizontal position slightly above the object sprayed. Stagnant regions within the booth should be carefully avoided or should be provided with a vertical exhaust. The air volume should be sufficient to maintain a velocity of 150 to 200 fpm over the open area of the booth and the vapors should be discharged through a suitable stack to permit dilution[4].

Hoods for Chemical Laboratories

Hoods used in chemical laboratories are generally provided with sliding windows which permit positive control of the fumes and vapors evolved by the apparatus. Their design should offer easy access for the installation of chemical equipment and should be well lighted. Air velocities should exceed 50 fpm when the window is opened to its maximum height.

[4]For a discussion of spray booths, see Special Bulletin No. 16, Spray Painting in Pennsylvania, Department of Labor and Industry, 1926, Harrisburg, Pa.

DESIGN OF DUCT SYSTEMS

The duct system should be large enough to transport the fumes or material without causing serious obstruction to the air flow. It is good practice to proportion the ducts to obtain the desired velocities and suction pressures at the hoods, although in many cases only an approximation to an ideal design is possible. Many exhaust hoods, and particularly those used in buffing and polishing, are connected by short branch pipes to the main duct which renders proportioning impractical.

Construction

The ducts leading from the hoods to the exhaust fan should be constructed of sheet metal not lighter than is shown in Table 4. The piping should be free from dents, fins and projections on which refuse might catch.

Table 4. Gage of Sheet Metal to be Used for Various Duct Diameters

Diameter of Duct	Gage of Metal
8 in. or less	24
9 to 18 in.	22
19 to 25 in.	20
26 in. or more	18

All permanent circular joints should be lap-jointed, riveted and soldered, and all longitudinal joints either grooved and locked or riveted and soldered. Circular laps should be in the direction of the flow, and piping installed out-of-doors should not have the longitudinal laps at the bottom. Every change in pipe size should be made with an eccentric taper flat on the bottom, the taper to be at least 5 in. long for each inch change in diameter. All pipes passing through roofs should be equipped with collars so arranged as to prevent water leaking into the building.

The main trunks and branch pipes should be as short and straight as possible, strongly supported, and with the dead ends capped to permit inspection and cleaning. All branch pipes should join the main at an acute angle, the junction being at the side or top and never at the bottom of the main. Branch pipes should not join the main pipes at points such that the material from one branch tends to enter the branch on the opposite side of the main.

Cleanout openings having suitable covers should be placed in the main and branch pipes so that every part of the system can be easily reached in case the system clogs. Either a large cleanout door should be placed in the main suction pipe near the fan inlet, or a detachable section of pipe, held in place by lug bands, may be provided.

Elbows should be made at least two gages heavier than straight pipe of the same diameter, the better to enable them to withstand the additional wear caused by changing the direction of flow. They should preferably have a throat radius of at least one and one-half times the diameter of the pipe.

Every pipe should be kept open and unobstructed throughout its entire

length, and no fixed screen should be placed in it, although the use of a trap at the junction of the hood and branch pipe is permissible, provided it is not allowed to fill up completely.

The passing of pipes through fire-walls should be avoided wherever possible, and sweep-up connections should be so arranged that foreign material cannot be easily introduced into them.

At the point of entrance of a branch pipe with the main duct, there should be an increase in the latter equal to their sum. Some state codes specify that the combined area be increased by 25 per cent. While this is not always necessary and is frequently done at the expense of a reduced air velocity, it is none the less advisable where future expansion of the exhaust system is contemplated.

TABLE 5. AIR SPEEDS IN DUCTS NECESSARY TO CONVEY VARIOUS MATERIALS

MATERIAL	AIR VELOCITIES (FPM)
Grain dust	2000
Wood chips and shavings	3000
Saw dust	2000
Jute dust	2000
Rubber dust	2000
Lint	1500
Metal dust (grindings)	2200
Lead dusts	5000
Brass turnings (fine)	4000
Fine coal	4000

Air Velocities in Ducts

When the static suction has been fixed for a given hood, the air velocity in the duct may be determined from Equation 2. Air velocities for conveying a material should be moderate. Table 5 gives the velocities generally employed for conveying various substances. Equations 5a and 5b may be used as tests to determine the conveying efficiency of a system. Velocities determined from these formulas should be increased by at least 25 per cent since they represent the minimum at which a stated size and density of material can be transported.

For vertical ducts:
$$V = 13{,}300 \frac{s}{s+1} d^{0.570} \tag{5a}$$

For horizontal ducts:
$$V = 6000 \frac{s}{s+1} d^{0.398} \tag{5b}$$

where

V = air velocity in duct, in feet per minute.
s = specific gravity of particles.
d = average diameter of largest particles conveyed, in inches.

Example 2. Granular material, the largest size of which is approximately 0.37 in. in diameter, with a specific gravity of 1.40 is to be conveyed in a vertical pipe the velocity of the air in which is 4100 fpm; find whether the material can be transported at this velocity.

Substitute data in Equation 5a and multiply by 1.25.

$$V = 1.25 \times 13{,}300 \times \frac{1.4}{2.4} \times 0.37^{0.57}$$

Antilog (0.57 × log 0.37) = 0.568; the required velocity is, therefore, 5500 fpm. Hence, the duct velocity must be increased either by speeding up the fan or decreasing the diameter of the duct or both.

Duct Resistance

The resistance to flow in any galvanized duct riveted and soldered at the joints may be obtained from Fig. 3, Chapter 19. The pressure drop through elbows depends upon the radius of the bend. For elbows whose centerline radii vary from 50 to 300 per cent of pipe diameter, the loss may be estimated from Table 6. It is sometimes convenient to express the resistance of an elbow in terms of an equivalent length of duct of the same diameter. Thus with a throat radius equal to the pipe diameter the resistance is equivalent to a section of straight pipe approximately 10 diameters long, while with a throat diameter radius 1½ times the diameter, the resistance is about the same as seven diameters of straight pipe.

COLLECTORS

The most common method of separating the dust and other materials from the air is to pass the mixture through a centrifugal or *cyclone* collector. In this type of collector the mixture of the air and material is introduced on a tangent, near the cylindrical top of the collector, and the whirling motion sets up a centrifugal action causing the comparatively heavy materials suspended in the air to be thrown against the side of the separator, from which position they spiral down to the tail piece, while the air escapes through the stack at the center of the collector.

The diameter of the cyclone should be at least 3½ times the diameter of the fan discharge duct. When two or more separate ducts enter a cyclone, gates should be provided to prevent any back draft through a system which may not be operating. Cyclones working in conjunction with two or more fans should be designed to operate efficiently at two-thirds capacity rating. The following formula is useful in computing the loss through a cyclone when the velocity of the air in the fan discharge duct is known:

$$h_c = 0.13 \left(\frac{V}{1000}\right)^2 \tag{6}$$

where

h_c = the pressure drop through the cyclone in inches of water.
V = the air velocity in the fan discharge duct in feet per minute.

If a cyclone is used to collect light dusts such as buffing wheel dusts, feathers and lint, the exhaust vent should be large enough to permit an air velocity of 200 to 500 fpm. This will, of course, require a cyclone of larger dimensions than given for the foregoing general case.

When a high collection efficiency is desired, or the material is very fine, multicyclones may be used. These are merely small cyclones arranged in parallel which utilize the principle of high centrifugal velocity to attain

separation. The capacities and characteristics of this type of separator should be obtained from the manufacturers.

Cloth Filters

Filter bags are used when the material collected by an exhaust system is valuable or cannot be separated from the air with an ordinary cyclone. They are also employed when it is desirable to recirculate the air drawn from a room by the exhaust system, which otherwise might entail considerable loss in heat. Bag filters which are properly housed may be operated under suction. *Bag houses* used in the manufacture of zinc oxide and other chemical products are operated on the positive side of the fan.

Wool, cotton and asbestos cloths are commonly used as filtering mediums. When woolen bags are employed, the filtering capacities vary from ½ to 10 cfm per square foot of filtering surface, depending on the character of the material collected. The rates for cotton and asbestos cloths are slightly lower. The type of filter cloth and the rates of filtration depend, of course, on the material to be collected and the fan capacity. The time increase of resistance varies with the amount of material permitted to build up on the surface of the filter and can only be determined by experiment. The limits of the increase may be regulated by adjustment of the shaking or cleaning mechanism. These limits may further be regulated according to the capacity of the fan and the effective performance of the hoods and the duct system.

RESISTANCE OF SYSTEM

The maintained resistance of the exhaust system is composed of three factors: (1) loss through the hoods, (2) collector drop, and (3) friction drop in the pipes.

The loss through the hoods is usually assumed to be equal to the suction maintained at the hoods. The collector drop in inches of water is given approximately by Equation 6, but where possible the resistance of the particular collector to be used should be ascertained from the manufacturer.

Friction drop in the pipes must be computed for each section where there is a change in area or in velocity. Find the velocities in each section of pipe starting with the branch most remote from the fan. The friction drop for these sections can be determined by reference to Table 6. Total friction loss in the piping system is the friction drop in the most remote branch plus the drop in the various sections of the main, plus the drop in the discharge pipe.

SELECTION OF FANS AND MOTORS

Manufacturers generally provide special fans for the collection of various industrial wastes. These are available for the collection of coal dust, wood shavings, wool, cotton and many other substances. For particular features concerning special fans, consult the Catalog Data Section of THE GUIDE and manufacturers' data. When substances having an abrasive character are conveyed, the fan blades and housing should be protected from wear. This may be accomplished by placing a

collector on the negative side of the fan or by lining the housing and blades with rubber.

If no future expansion of an exhaust system is contemplated, the fan motor should be chosen to provide the calculated air volume. Should, however, the exhaust system be required to handle more air in the future, the motor should be adequate for the maximum load anticipated. Further information regarding the choice of fans and motors is given in Chapter 17.

PROTECTION AGAINST CORROSION

The removal of gases and fumes in many chemical plants requires that metals used in the construction of the exhaust system be resistant to chemical corrosion. A list of the materials which may be used to resist the action of certain fumes is given in Table 7. Hoods and ducts when short, may frequently be constructed of wood and be quite effective.

Table 6. Loss Through 90-Deg Elbows

Elbow Center Line Radius in Per Cent of Pipe Diameter	Loss in Per Cent of Velocity Head
50	75
100	26
150	17
200 to 300	14

Table 7. Materials to be Used for the Protection of Exhaust Systems Against Corrosion[a]

Type of Fume Conveyed	Protective Material to be Used
Chlorine	Rubber lining or chrome-nickle alloys
Hydrogen sulphide	Aluminum coated iron, aluminum, high chrome-nickle alloys
Ammonia	Iron or steel
Sulphurous gases	High chrome-nickle alloys
Hydrocholric acid	Rubber lining, chrome-nickle alloys
Nitrous gases	Nickle-chrome alloys

[a]Condensed from data given by Chilton and Huey (*Industrial and Engineering Chemistry*, Vol. 24, 1932).

Rubberized paints are available and may be applied as protective coatings in handling such gases and fumes as chlorine and hydrochloric acid.

Chapter 22

CENTRAL FAN HEATING SYSTEMS

Types of Systems, Blow-Through, Draw-Through, Heating Units, Design, Temperatures, Weight of Air to be Circulated, Temperature Loss in Ducts, Heat Supplied Heating Units and Washer, Grate Area, Boiler Selection, Weight of Condensate, Static Pressure, Fans and Control

A CENTRAL fan system is an indirect system of heating, ventilating, or air conditioning in which fans and blowers distribute air through ducts from one centrally located plant. This chapter considers heating and humidifying systems of this type; similar systems arranged for cooling and dehumidifying are discussed in Chapter 9. A special type of central fan system, the mechanical warm air or fan furnace system, which is especially adapted to residences and other small buildings is covered in Chapter 23.

TYPES OF SYSTEMS

In the indirect type of central fan heating and air conditioning system, steam is usually the medium by which heat is transferred from the boiler, or other source of heat, to the heating units. If the system is intended solely for heating, the air is passed over one or more stacks or batteries of heating units and then conveyed to the spaces for which it is intended through a system of ducts. In some cases, a predetermined amount of outside air is introduced for ventilating purposes, whereas in others the moisture content is controlled by passing the air through a washer or humidifier. If the apparatus is designed to control simultaneously the temperature, humidity, air motion, and distribution, it is known as an air conditioning system.

In the *split system*, the heating is accomplished by means of radiators or convectors, and the ventilating or air conditioning by means of the central fan apparatus. In the *combined system*, the entire operation of heating, ventilating, and air conditioning is handled by the central fan system.

A common arrangement of the central fan system of heating is illustrated by Fig. 1 and consists of a fan, heating unit (heater) enclosed by a sheet metal casing connected with the suction side of the fan, a sheet metal casing connected to the heating unit casing run to the outside of the building and provided with an adjustable opening inside the building for recirculation of the air when desired, and a duct system attached to the fan outlet to convey and distribute the air to various parts of the building to be warmed by the apparatus. The fan is ordinarily motor-driven; there are, however, many cases when a direct-connected steam engine may be used to advantage. In this event the exhaust from the engine can be con-

nected to one or more sections of the heater, depending upon the condensation rate of the engine. The recirculation duct connected with the opening in the suction duct should be extended to a point as near the floor as possible.

When ventilation is not a requirement or is considered relatively unimportant, as in shop and factory heating, and the number of persons vitiating the air is small compared with the cubical contents of the building, or the process does not generate obnoxious gas or vapors, the air may be recirculated, sufficient outside air for ventilation being supplied by infiltra-

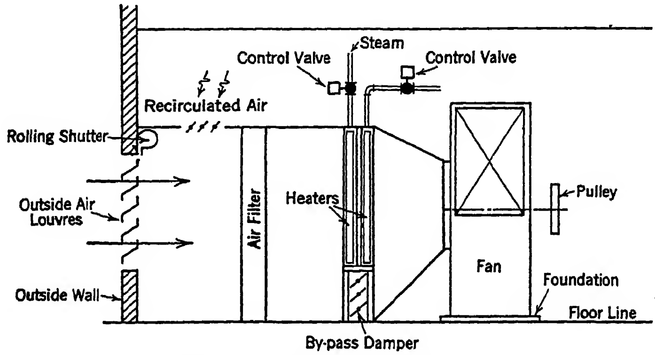

FIG. 1. ARRANGEMENT OF A CENTRAL FAN HEATING SYSTEM. (DRAW-THROUGH)

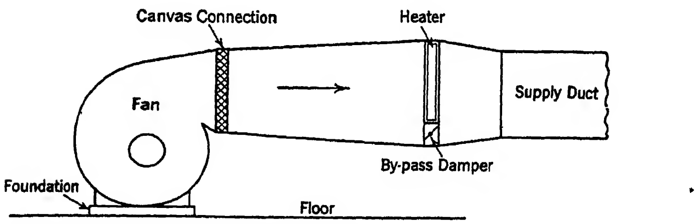

FIG. 2. ARRANGEMENT FOR HEATING UNIT AND BY-PASS. (BLOW-THROUGH)

tion. The amount of heat to be supplied the heating unit in this case is the same as would be required for a direct radiation installation.

When ventilation is a requirement to be met, an arrangement similar to that shown by Fig. 1 may be employed. Since the amount of air necessary for heating is generally in excess of the amount required for ventilation, considerable fuel economy may be effected by recirculating a portion of the air. In this case only sufficient outside air is drawn into the system to meet the ventilation requirement and the remainder of the air, required for heating, is recirculated. This may be readily effected by an arrangement of ducts and dampers on the suction side of the fan as previously mentioned. If the outside air introduced is to be washed or conditioned the washer or humidifier and tempering coil may be added between the inlet for the recirculated air and the fresh air intake.

Blow-Through, Draw-Through

When the heating unit is located on the suction side of the fan, the system is known as *draw-through.* (See Fig. 1). When the heating unit is located in the discharge from the fan the system is known as *blow-through.* (See Fig. 2). The draw-through combination is used for factory and toilet room installations because a more compact arrangement of the apparatus usually is possible. In addition, air leakage will be inward. The blow-through combination is used principally in schools and public buildings, and for all booster coil arrangements where different temperatures and independent temperature regulation are required for different heated spaces. In public building installations, the fan frequently blows the heated air into a plenum chamber from which the air ducts radiate to the various rooms of the building; this arrangement is sometimes called the *plenum* system.

HEATING UNITS

The heating units for central fan systems using steam as the heating medium may be classified as (1) tempering coils, (2) preheater coils, (3) reheater coils, (4) booster coils, and (5) water heaters, either open or closed. *Tempering coils* are used with ventilating and air conditioning systems for raising the temperature of the outside cold air to above freezing, or 32 F. They are not required for heating systems where all of the air is recirculated, since the temperature of the recirculated air will be above freezing. *Preheater coils* are used with air conditioning systems to raise the temperature of the air from that leaving the tempering coils to such a temperature that in passing through the water sprays of the washer (without water heater) the air will become partially saturated (adiabatically) having a moisture content corresponding to the required dew-point temperature. Preheater coils therefore supply heat as necessary to control the dew-point temperature. The *reheater coils* are used to raise the temperature of the air leaving the tempering coils (in the case of a heating or ventilating system) or the air leaving the washer (in the case of an air conditioning system) to that necessary to maintain the desired temperature in the rooms or spaces to be heated or conditioned, except where booster coils are used in which case the reheater coils raise the air temperature to approximately room temperature, or slightly higher. *Booster coils* are installed in the duct branches to control the temperature of the air entering the rooms or spaces for which it is intended. *Water heaters* are used on an air conditioning system to control the dew-point temperature. They are used mainly for industrial work, seldom for comfort conditioning. They are not used where preheater coils are employed. The open type supplies steam directly to the spray water, while the closed type utilizes a heat interchanger by which the steam imparts its heat to the spray water. Where water heaters are required for comfort conditioning, the closed type is used.

The heating units for central fan systems in use at the present time consist either of pipe coils, finned tubes of steel, copper, brass or other metal, cast-iron sections with extended surfaces, or the cellular type. Steam is passed through these heating units and the air to be heated is passed over their exterior surfaces.

In selecting a heating unit for any particular service, the choice should be based on the desired requirements as follows:

1. Final temperature desired.
2. Loss in pressure (or friction) of air passing over the heating unit.
3. Air velocity over the heating unit.
4. Free area or face area of heating unit.
5. Ratio of heating surface to net free (or face) area.
6. Air volume required.
7. Rows deep of pipe, tubes or sections.
8. Amount of heating surface.
9. Steam pressure drop through the heating unit.
10. Weight of heating unit.

Final Temperature Desired. The choice of a heating unit is largely influenced by the final temperature desired, when the entering air temperature and steam pressure available at the heating unit are specified. These data are obtainable from manufacturers' catalogs.

Loss in Air Pressure (or Friction). The allowable friction through the heating unit is one of the first factors to be determined in the selection of the apparatus. The velocities of air through various types of heating units will not necessarily be the same, but for any particular job the velocity through the heating unit should be a secondary consideration and the allowable friction or air pressure loss should be fixed approximately before proceeding with the selection of the heating unit. The loss in air pressure (or friction) through the heating unit should not exceed a predetermined maximum allowable amount for economical operation and for moderate size and first cost of installation.

In public building work, the maximum allowable friction through both tempering coil and reheater coils should never exceed ⅝ in. of water and it is advisable that the friction be kept considerably lower than this figure if possible. A tempering coil friction ranging from 0.10 to 0.20 in. of water is considered satisfactory. The air pressure loss for reheaters ordinarily ranges from 0.20 to 0.40 in. of water. In factory work, the maximum friction through the heater should never exceed 0.8 in. or 1 in. of water and it is advisable to figure the heaters at lower frictions if possible.

Velocity through Heating Unit. This velocity has generally been given in manufacturers' tables as being measured at 70 F and in most cases refers to the velocity through the net free area of the heating unit, or through the net space between the pipes, tubes or sections. Although most manufacturers give suitable velocities measured at 70 F, certain manufacturers show velocities measured at 65 F and others indicate velocities measured at the average air temperature through the heating unit. Many new heating units, however, specify net face areas with corresponding velocities instead of velocities through net free areas. In either case, manufacturers publish the corresponding friction or air-pressure loss in tables. The velocity through the net free area of the heating unit averages about 1000 fpm and that through the net face area about 500 fpm.

The *volume of air* to be heated in any particular case is determined after consideration of the ventilation requirements, heat losses, and quantity of air required for proper circulation, as explained in Chapters 2 and 7.

The number of *rows of pipe tubes or sections* or the amount of *heating surface* to be used may be selected from manufacturers' catalogs after the quantity of air handled and heat load are known. Savings in operating expense or cost of installation should result from a proper selection of heater and by-pass areas. For example, instead of having the entire air quantity go through a one-row heating unit, it may be advantageous to use a two-row heating unit and a properly sized by-pass. Thus, when no heating is being done, a suitable by-pass damper may be opened to place a lighter load on the fan.

The *steam pressure drop through the heating unit* is also tabulated in manufacturers' data tables. The sizing of steam supply and return piping, allowing for drops through heating units, is explained in Chapter 32.

Weight of Heating Unit. In the design of a heating system, the weight limitations of heating units are determined by the location of the units. Obviously, if there is no loading limitation imposed, any type of heating unit may be selected. On the other hand if the heating unit is to be hung from the ceiling, it may be desirable to use the lightest unit which will accomplish the work required.

DESIGNING THE SYSTEM

The general procedure for the design of central fan systems is as follows:

1. Calculate the heat loss for each room or space to be heated.
2. Determine volume of outside air to be introduced.
3. Assume or calculate temperature of air leaving registers or supply outlets.
4. Calculate weight of air to be circulated.
5. Estimate temperature loss in duct system.
6. Calculate heat to be supplied the heating units and washer.
7. Select heating units and washer from manufacturers' data and performance curves.
8. Calculate total heat to be supplied.
9. Calculate grate area and select boiler.
10. Design duct system.
11. Calculate total static pressure of system.
12. Select fan, motor, and drive.

The heat losses (H) should be calculated in accordance with the procedure outlined in Chapter 7. If a positive pressure is maintained by the central fan system in the room or space to be ventilated or conditioned, there will ordinarily be very little infiltration of cold outside air through the cracks and crevices of the space. Consequently, the volume of air introduced into the space at the assumed or calculated outlet temperature need only be sufficient to provide for the transmission losses, plus a part of the infiltration losses, about one-third. The exfiltration of heated or conditioned air through the cracks and crevices of the space should be provided for by making the usual allowance for the infiltration losses in arriving at the total heat loss of the space. The air required to make up for this exfiltration of heated or conditioned air will be brought in at the outside air intake and may be included as a part of the outside air necessary for the ventilating requirements. The heat required to raise this air to the

conditions maintained in the room must be provided by the tempering coils, preheater coils and reheater coils. If a positive pressure is not maintained in the room or space to be conditioned, the normal infiltration of outside cold air will take place in this room, and the outlet temperature, together with the required air volume at this temperature, must be sufficient to provide for both the infiltration and transmission losses.

Volume of Outside Air

The volume of outside air required for ventilation or air conditioning purposes may be determined from data in Chapter 2. In no case shall less than 10 cfm per person be introduced.

The heat required to warm the outside air introduced for ventilation purposes (H_o) may be determined by means of the following formula:

$$H_o = 0.24\ (t - t_o)\ M_o \tag{1}$$

where

0.24 = specific heat of air at constant pressure.
t = room temperature, degrees Fahrenheit.
t_o = outside temperature, degrees Fahrenheit.
M_o = weight of outside air to be introduced per hour in pounds = d_oQ_o.
Q_o = volume of outside air to be introduced, cubic feet per hour.
d_o = density of air at t_o, pounds per cubic foot.

Example 1. A building in which the temperature to be maintained is 70 F requires 10,000 cfm. If the outside temperature is 20 F, how much heat will be required to warm the air introduced for ventilation purposes to the room temperature?

Solution. Q_o = 10,000 × 60 = 600,000 cfh; d_o = 0.08276 (Table 1, Chapter 41); M_o = 0.08276 × 600,000 = 49,656 lb; t = 70 F; t_o = 20 F; H_o = 0.24 × (70 − 20) × 49,656 = 595,872 Btu per hour.

Temperature of Air Leaving Registers

If the system is to function only as a heating system, that is, entirely as a recirculating one, the temperature of the air leaving the register outlets must be assumed. For public buildings, these temperatures may range from 100 to 120 F, whereas for factories and industrial buildings the outlet or register temperature may be as high as 140 F. In no case should the outlet temperature exceed these values.

For ventilating or conditioning systems, the temperature of the air leaving the supply outlets may be estimated by means of the following formula:

$$t_y = \frac{55.2H}{Q} + t \tag{2}$$

where

t_y = outlet temperature, degrees Fahrenheit.
H = heat loss of room or space to be conditioned, Btu per hour.
Q = *total* volume of air to be introduced at the temperature, t, cubic feet per hour.

If the outlet temperature (t_y) as determined from Equation 2 exceeds 120 F for public buildings, or 140 F for factories or industrial buildings, this temperature should be assumed using the maximum permissible

temperature and the volume of air to be introduced into the room or space determined by means of the following equation:

$$Q = \frac{55.2H}{(t_y - t)} \tag{3}$$

Example 2. The heat loss of a certain auditorium to be conditioned is 100,000 Btu per hour. The ventilating requirements are 90,000 cu ft per hour and the room temperature 70 F. Determine the outlet temperature.

Solution. Substituting in Formula 2,

$$t_y = \frac{55.2 \times 100{,}000}{90{,}000} + 70 = 131.3 \text{ F}$$

Inasmuch as this temperature is excessive, it will be necessary to assume the outlet temperature, which will be taken as 120 F, and to calculate the amount of air to be introduced into the room at this temperature to provide for the heat loss. Substituting in Equation 3,

$$Q = \frac{55.2 \times 100{,}000}{120 - 70} = 110{,}400 \text{ cfh (at temperature } t)$$

Weight of Air to be Circulated

The total weight of air to be introduced into the room or space to be heated or conditioned (M) is given by the following formulae:

$$M = \frac{H}{0.24(t_y - t)} = dQ \tag{4}$$

$$M = M_o + M_r \tag{5}$$

$$M_o = d_oQ_o \tag{6}$$

where

d = density of air at temperature t, pounds per cubic foot.
d_o = density of air at temperature t_o, pounds per cubic foot.
Q_o = volume of outside air at temperature t_o.
M_o = weight of outside air, pounds.
M_r = weight of recirculated air, pounds.

Example 3. Using the data of Example 2 and an outside temperature of 20 F, what will be the values of M, M_o and M_r?

Solution. $d = 0.07495$, $d_o = 0.08276$, $Q = 110{,}400$, $Q_o = 90{,}000$, $H = 100{,}000$.

$$M = \frac{100{,}000}{0.24 \times (120 - 70)} = 8{,}333 \text{ lb}$$

$$M_o = 0.08276 \times 90{,}000 = 7{,}448 \text{ lb}$$

$$M_r = M - M_o = 8{,}333 - 7{,}448 = 885 \text{ lb}$$

Temperature Loss in Ducts

The allowances to be made for loss in transit through the duct system (t_z) are as follows:

1. When the duct system is located in the enclosure to which the air is being delivered, as in a factory, it may be assumed that there is no loss between the reheater coil and the point or points of discharge into the enclosure.

2. For ducts in outside walls or attics, or other exposed places, allow 0.25 deg per linear foot of uninsulated duct.

3. For ducts run underground an allowance shall be made based on the estimated heat loss of the duct, assuming an average temperature of the ground of 55 F.

Heat Supplied Heating Units and Washer

The following cases may arise in practice:

A. The heating of the building is done entirely by means of a central fan system, all of the air being drawn from the outside.

B. Similar to (*A*), except that all of the air is recirculated.

C. A portion of the air is recirculated, and the remainder is drawn in from the outside.

D. Air at the same temperature to be delivered to all the rooms. A constant relative humidity is maintained in the building and all of the air circulated is drawn from outside the building. (Not applicable to the heating of various rooms where individual control of each room is desired).

E. A system using outside air, return air, and by-pass air, reheater being located in by-pass air chamber.

F. Arrangement of apparatus where individual control of the temperature for each room is required in conjunction with air washer equipment to maintain a constant relative humidity in the rooms. The air washer is provided with a water heater for the spray water, capable of fully saturating the air. A section of preheater may be used for this purpose in place of the water heater. With this arrangement and with a uniform temperature of air entering the rooms, it is impossible to maintain the same room temperature throughout the building because the weight of air to be delivered to each room is determined and fixed by the ventilating requirements.

In analyzing these cases, the following symbols will be used:

H = heat loss of the room or building, Btu per hour.

H_1 = heat to be supplied to the reheater coil, Btu per hour.

H_2 = heat supplied tempering coil, or tempering coil and preheater, Btu per hour.

H_3 = heat supplied air washer by water heater, Btu per hour.

H_4 = heat to be supplied booster coil, Btu per hour.

M = weight of air to be introduced into the room or building, pounds per hour.

M_r = weight of recirculated air, pounds per hour.

M_b = weight of air by-passing washer, pounds per hour.

M_o = weight of air drawn in from outside, pounds per hour.

t_o = mean temperature of outside air, degrees Fahrenheit.

t = mean air temperature to be maintained in the room or building, degrees Fahrenheit.

t_1 = mean temperature of the air entering the reheater coil.

t_2 = mean temperature of the air leaving the reheater coil.

t_z = temperature loss in the duct system.

t_y = temperature of the air leaving the duct outlets.

t_x = average temperature of air entering tempering coil.

t_w = temperature of air entering washer.

0.24 = specific heat of air at constant pressure.

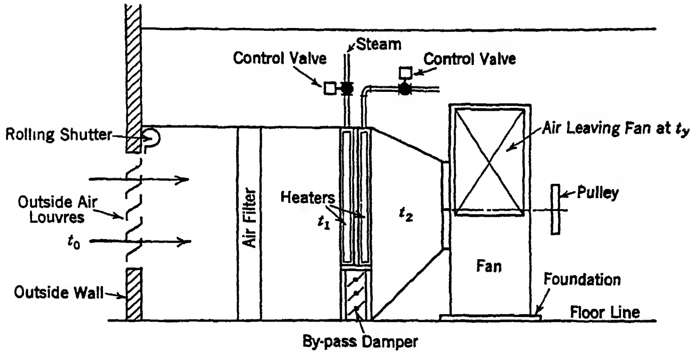

FIG. 3. HEATING UNIT AND FAN ARRANGED FOR OUTSIDE AIR CIRCULATION. (*Case A*)

Case A. (Fig. 3). All of the air circulated to be drawn from outside the building, in which case $t_x = t_o$

$$H_2 = 0.24\ (t_1 - t_o)\ M_o \tag{7}$$

$$H_1 = 0.24\ (t_2 - t_1)\ M_o \tag{8}$$

Example 4. The heat loss H for a certain factory building is 700,000 Btu per hour. The mean inside temperature t to be maintained is 65 F. The assumed outside air temperature t_o is 0 deg; $t_x = 0$, $t_y = t_2$ and is assumed to be 140 F. The temperature leaving the tempering coil is assumed to be 35 F. Required, H_1 and H_2. From Equation 4,

$$M = \frac{700{,}000}{0.24\ (140 - 65)} = 38{,}889 \text{ lb per hour.}$$

$H_2 = 0.24 \times (35 - 0) \times 38{,}889 = 326{,}667$ Btu per hour.
$H_1 = 0.24 \times (140 - 35) \times 38{,}889 = 980{,}003$ Btu per hour.
$H_2 + H_1 = 326{,}667 + 980{,}003 = 1{,}306{,}670$ Btu per hour.

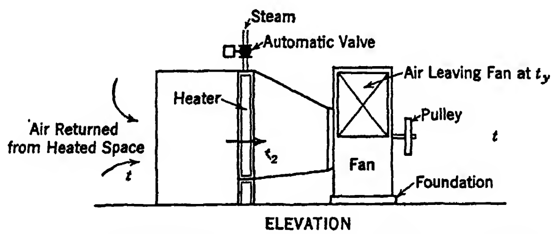

FIG. 4. ARRANGEMENT FOR RECIRCULATION. (*Case B*)

Case B. (Fig. 4) All of the air is to be recirculated, in which case $t_1 = t$.

$M_r = 38{,}889$ lb
$M_1 = 0.24\ (t_2 - t_1)\ M_r$
$H_1 = 0.24\ (140 - 65) \times 38{,}889 = 700{,}000$ Btu per hour.

This example illustrates the saving in fuel consumption by the recirculation of the air. The heat to be supplied the apparatus is the same as that required for a direct system of heating and is equal to the heat loss of the building ($H_1 = H$), in the example 700,000 Btu per hour as compared with 1,306,670 for *Case A*.

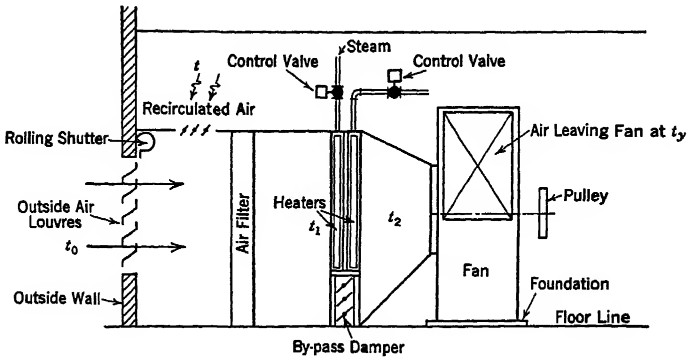

FIG. 5. COMBINATION OF RECIRCULATED AIR AND OUTSIDE AIR. (*Case C*)

Case C. (Fig. 5) A portion of the air circulated is recirculated air and the remainder, as may be required for ventilating purposes, is drawn in from the outside. According to Equations 4 and 5,

$$M = M_o + M_r = \frac{H}{0.24\ (t_y - t)}$$

The temperature of the resulting mixture of outside and recirculated air entering the tempering coil is:

$$t_x = \frac{M_o t_o + M_r t}{M} \qquad (9)$$

Example 5. Assuming that a positive supply of outside air (d_o = 0.0864) is required for ventilation at the rate of 90,000 cu ft per hour in the preceding example, then M_o = 0.0864 × 90,000 = 7776 lb per hour are required, measured at 65 F.

$$M_r = M - M_o = 38{,}889 - 7776 = 31{,}113 \text{ lb.}$$

$$t_x = \frac{7776 \times 0 + 31{,}113 \times 65}{38{,}889} = 52 \text{ F}$$

$$H_1 = 38{,}889 \times 0.24\ (140 - 52) = 821{,}336 \text{ Btu.}$$

This amount of work may be accomplished with one or more banks of heating units, that is, either a single reheater or a tempering coil and reheater.

The three preceding cases refer to installations in which conditioning the air to maintain certain relative humidity requirements does not enter into the problem, as for example, certain types of industrial installations. In practically all modern public buildings, theaters, schools, and in many industrial installations the ventilating requirements include the provision for air washing and humidifying the air delivered to the various rooms of the structure.

In the following cases it is assumed that in addition to maintaining a mean room temperature t, the heating and ventilating apparatus is required to maintain a constant relative humidity in the rooms.

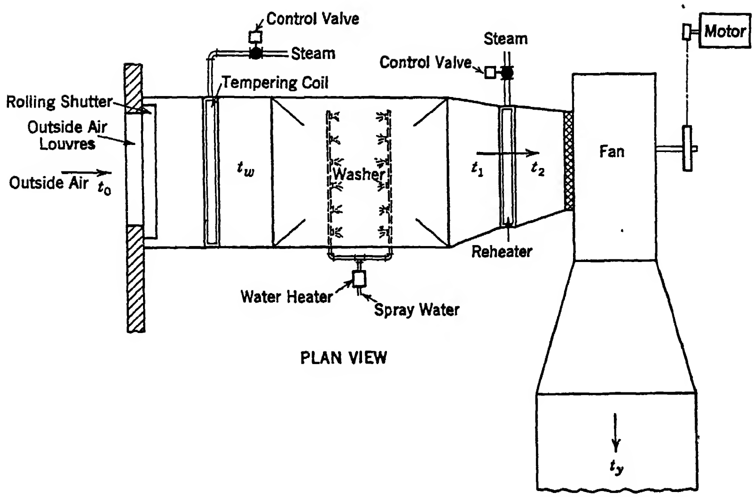

FIG. 6. OUTSIDE AIR CIRCULATED; CONSTANT RELATIVE HUMIDITY IN ROOM. (*Case D*)

Case D. (Fig. 6) The maximum relative humidity that may be maintained within the building without the precipitation of moisture on single glazed sash when the outside temperature is 30 F is approximately 35 per cent. If the inside temperature t is 70 F, 35 per cent relative humidity corresponds to a dew-point temperature of 41 F. (See psychrometric chart, Chapter 1).

The installation shown in Fig. 6 contemplates the use of a tempering coil, air washer provided with a water heater, and a reheater. The tempering coil, one section in depth, warms the incoming air to approximately 35 F to prevent freezing any of the spray water. The air passing through the spray chamber is saturated and leaves at a temperature of $t_1 = 41$ F.

The heat to be supplied the reheater is:

$$H_1 = 0.24\ (t_2 - 41)M \text{ Btu per hour.}$$

The heat to be supplied the tempering coil is:

$$H_2 = 0.24\ (35 - t_0)M \text{ Btu per hour.}$$

The amount of heat, per pound of air circulated, to be supplied the humidifying washer or humidifier is the difference between the heat content of the assumed dry air entering the washer at a temperature of $t_W = 35$ F and the leaving saturated air at $t_1 = 41$ F (Chapter 1):

$$15.7 - 8.4 = 7.3 \text{ Btu per pound of dry air.}$$

The amount of heat required for the washer is:

$$H_3 = 7.3\ M \text{ Btu per hour.}$$

The total amount of heat required by the apparatus is therefore;

$$H_1 + H_2 + H_3 \text{ Btu per hour.}$$

If a washer having a *humidifying efficiency* of 67 per cent *without water heater* is employed it will be necessary to heat the outside air drawn into the apparatus by means of the tempering and preheater coils to such a temperature that the air in passing through

the water sprays will become partially saturated (adiabatically) having a moisture content per pound of air equal to saturated air at 41 F. If the incoming air is warmed to t_w = 88 F (requiring a two-section-depth heating unit) it will be cooled in the washer to 64 F or 88 − 64 = 24 deg.

If the *humidifying efficiency* of the washer were 100 per cent, the air would become adiabatically saturated at 52 F or a temperature drop of 88 − 52 = 30 F. The efficiency of the washer is, however, only 67 per cent, so that the actual temperature drop will be 0.67 × 36 deg or 24 deg, as used.

The heat to be supplied the reheater is in this case $H_1 = 0.24\ (t_2 - 64)M$ Btu per hour, and for the tempering coil and preheater is $H_2 = 0.24\ (88 - t_o)M$. The total heat required by the apparatus is $H_1 + H_2$, no heat being supplied to the washer.

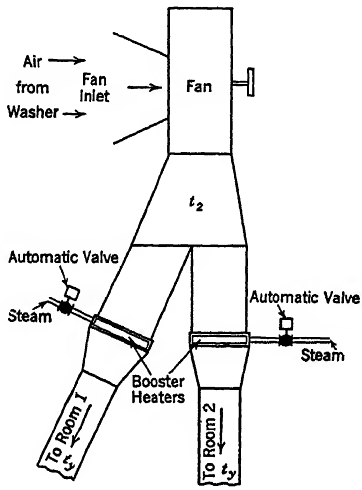

FIG. 7. OUTSIDE AIR CIRCULATED; CONSTANT TEMPERATURE AND RELATIVE HUMIDITY MAINTAINED IN EACH ROOM. (*Case E*)

Case E. (Fig. 7). The temperature t_y will ordinarily be different for each room With H and M fixed, $0.24\ (t_y - t)M = H$, or

$$t_y = \frac{H}{0.24\ M} + t$$

In order to provide the proper temperature for each room, a booster coil is generally installed in each supply duct near the outlet to control the outlet temperature t_y. The amount of steam supplied to these booster units is usually controlled automatically by individual thermostats. The heat required by the booster coils depends on the temperature range through which the air is heated and the quantity of air, or

$$H_4 = 0.24\ (t_y - t_2 - t_z)M \tag{10}$$

Total Heat to be Supplied

The total heat to be supplied (H^1) is equal to the sum of the heat requirements of the various heating units and the water heater of the washer, if any, plus the allowance for piping tax, etc. (See preceding *Cases A* to *E*.

Grate Area, Boiler Selection

The required grate area may be determined by the following formula:

$$G = \frac{H^1}{F \times E \times C} \tag{11}$$

where

G = required grate area, square feet.
F = calorific value of fuel, Btu per pound.
C = combustion rate, pounds per square foot of grate per hour.
E = boiler and grate efficiency, per cent.

Example 6. Using the data in Example 4, and assuming coal having a calorific value of 12,000 Btu per pound, a combustion rate of 7 lb per square foot, and a performance efficiency of 0.60, and neglecting piping tax, etc.,

$$G = \frac{1{,}306{,}670}{12{,}000 \times 0.60 \times 7} = 26 \text{ sq ft}$$

Weight of Condensate

The normal weight of condensate to be handled from central fan systems may be estimated by means of the following formula:

$$W = \frac{60 \times Q \times \Delta t}{55.2 \times h_{fg}} \tag{12}$$

where

W = weight of condensate, pounds per hour.
Q = total volume of air, cubic feet per minute.
Δt = temperature rise of air, degrees Fahrenheit.
h_{fg} = latent heat of steam in the system, Btu per pound.

Ducts and Outlets, Air Filters, Air Washers

The design of the duct system should be based on data contained in Chapter 19. Air washers and humidifiers are described in Chapter 11. For information on air filters, see Chapter 16.

Static Pressure

The total static pressure against which the system must operate may be found by summing up the static losses through the complete system from the outside air intake to the discharge outlets or nozzles. This means that the loss due to friction must be determined for each piece of apparatus involved. Most of these values may be obtained from manufacturers' data tables. For a simple system, the following static pressure drops may be assumed:

1. Outside air inlet, comprised of screen, louver and short duct, may have a loss of 0.2 in. of water.
2. A typical oil filter at rated capacity and velocity has a drop of 0.25 in. of water.
3. The loss of one row of a standard make tempering stack equals 0.09 in. water.
4. The loss of one row of a standard make preheater equals 0.10 in. water.
5. A standard humidifier at rated velocity may have a loss of about 0.35 in. water.
6. The loss through one row of a standard make reheater equals 0.12 in. water.
7. A fair assumption for duct losses on a simple system is 0.25 in. water.
8. The static pressure for a nozzle type outlet may be taken as 0.1 in. water.

The sum of these values equals 0.2 + 0.25 + 0.09 + 0.10 + 0.35 + 0.12 + 0.25 + 0.1 = 1.46 in. which is the static pressure against which the system must operate.

Fans and Control

The selection of fans and motors may be based on data contained in Chapter 17. Because centrifugal fans reach their maximum efficiency when working against the resistance offered by the average central fan heating system, they are well adapted to such systems and are generally used. Information on temperature control for central fan systems is given in Chapter 14.

Chapter 23

MECHANICAL WARM AIR AND FAN FURNACE SYSTEMS

Fan Furnaces, Fans and Motors, Elimination of Noise, Air Washers and Filters, Cooling Methods, Duct Design, Controls, Selecting the Furnace, Selecting the Fan, Provision for Cooling System, Heavy Duty Fan Furnaces

MECHANICAL warm air or fan furnace heating systems, which are a special type of central fan systems, are particularly adapted to residences, small office buildings, stores, banks, schools, and churches. Circulation of air is effected by motor-driven fans instead of by the difference in weight between the heated air leaving the top of the casing and the cooled air entering its bottom, as in gravity systems described in Chapter 24. The advantages of mechanical systems, as compared with gravity systems, are as follows:

1. Furnace can be installed in a corner of the basement, leaving more basement room available for other purposes.
2. Basement distribution piping can be made smaller and can be so installed as to give full head room in all parts of the average basement, or be completely concealed from view except in the furnace room.
3. Circulation of air is positive, and in a properly designed system can be balanced in such a way as to give a greater uniformity of temperature distribution.
4. Humidity control is more readily attained.
5. The air may be cleaned by air washers or filters, or both.
6. Some cooling effect in summer will result from the installation of a properly designed system.
7. The fan and duct equipment may be utilized for a complete cooling and dehumidifying system for summer, using either ice, mechanical refrigeration, or low temperature water for cooling and dehumidifying, or adsorbers for dehumidifying.
8. The use of the fan increases the volume of air which can be handled, thereby increasing the rate of heat extraction from a given amount of heating surface and insuring sufficient air volume to obtain proper distribution in a large room.

Much of the equipment used in central fan systems is the subject matter of other chapters. It is the purpose of this chapter to discuss the coordinated design and to deal in detail only with problems not covered elsewhere which refer particularly to the whole problem of fan warm air furnace heating and air conditioning.

FAN FURNACES

Furnaces for mechanical warm air systems may be made of cast-iron, steel, or alloy. Cast-iron furnaces are usually made in sections and must be assembled and cemented or bolted together on the job. Steel furnaces

are made with welded or riveted seams. The proper design of the furnace depends largely on the kind of fuel to be burned. Accordingly, various manufacturers are making special units for coal, oil and gas. Each type of fuel requires a distinct type of furnace for highest efficiency and economy, substantially as follows:

1. Coal Burning:
   a. Bituminous—Large combustion space with easily accessible secondary radiator or flue travel.
   b. Anthracite or coke—Large fire box capacity and liberal secondary heating surfaces.
2. Oil Burning:
   a. Liberal combustion space.
   b. Long fire travel and extensive heating surface.
3. Gas Burning:
   a. Extensive heating surface.
   b. Close contact between flame and heating surface.

A combustion rate of from 5 to 8 lb of coal per square foot of grate per hour is recommended for residential heaters. A higher combustion rate is

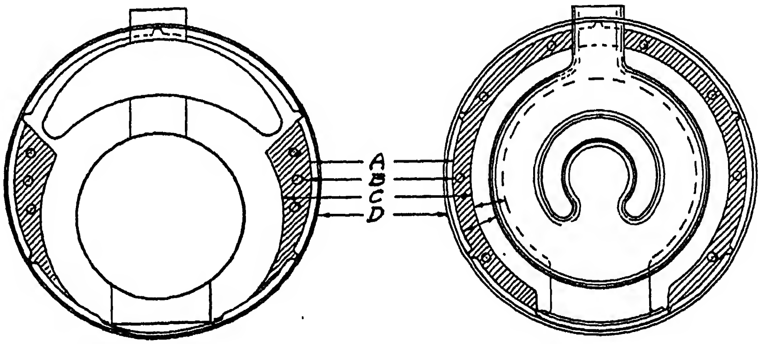

FIG. 1. USUAL METHOD OF BAFFLING ROUND CASINGS FOR FAN FURNACE WORK

A. Liner, 1 in. from casing. *B.* Hole to vent baffle.
C. Baffle, closed top and bottom. *D.* Outer casing.

permissible with larger furnaces for buildings other than residences, depending upon the ratio of grate surface to heating surface, firing period, and available draft.

Where oil fuel is used, care must be exercised in selecting the proper size and type of burner for the particular size and type of furnace used. It is recommended, especially where sectional furnaces are used, that the system be designed for blow-through installations, so that the furnace shall be under external pressure, in order to minimize the possibility of leakage of the products of combustion into the air circulating system.

In residential furnaces for coal burning, the ratio of heating surface to grate area will average about 20 to 1; in commercial sizes it may run as high as 50 to 1, depending on fuel and draft. Furnaces may be installed singly, each furnace with its own fan, or in batteries of any number of furnaces, using one or more fans.

Casings are usually constructed of galvanized iron, 26-gage or heavier,

but they may also be constructed of brick. Galvanized-iron casings should be lined with black-iron liners, extending from the grate level to the top of the furnace and spaced from 1 in. to 1½ in. from the outer casing. It is generally believed that either brick or sheet metal casings should be equipped with baffles to secure impingement of the air to be heated against the heating surfaces. Brick furnace casings should be supplied with access doors for inspection.

Most furnace casings are sized for gravity flow of air, and where a fan is to be used, many manufacturers recommend the use of special baffles to restrict the free area within the casing and to force impingement of the air against the heating surfaces. The method of making these baffles for furnaces with top horse-shoe radiators and for furnaces with back crescent radiators is illustrated in Fig. 1.

Either square or round casings may be used. Where square casings are used, the corners must be baffled to reduce the net free area and to force impingement of air against the heating surfaces. Fig. 2 shows the usual method of baffling square furnace casings for fan-furnace work.

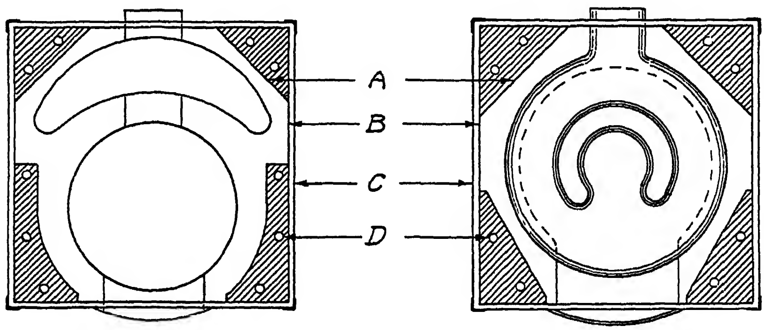

Fig. 2. Method of Baffling Square Furnace Casing for Fan Furnace Work

A. Baffle, closed top and bottom. *B*. Liner, 1 in. from casing. *C*. Outer casing. *D*. Hole to vent baffle.

The hood or bonnet of the casing above the furnace should be as high as basement conditions will allow, to form a plenum chamber over the top of the furnace. This tends to equalize the pressure and temperature of the air leaving the bonnet through the various openings. It is generally considered advisable to take off the warm air pipes from the side of the bonnet near the top, as this method of take-off allows the use of a higher bonnet and thus provides a larger plenum chamber. Fig. 3 illustrates a complete fan-furnace installation showing location of fan, furnace, filters, plenum chamber and method of take-off of warm air pipe.

FANS AND MOTORS

Centrifugal type fans are most commonly used, and these may be equipped with either backward or forward curved blades. Low tip speed is desirable for the elimination of air noise, especially where forward curved blades are used. Motors may be mounted on the fan shaft or outside of the fan with belt connection. Multi-speed motors or pulleys

are desirable to provide a factor of safety and to allow for more rapid circulation for summer cooling.

For additional information on fans and motors, see Chapter 17.

ELIMINATION OF NOISE

Special attention must be given to the problem of noise elimination. The fan housing must not be directly connected with metal, either to the furnace casing or to the return air piping. It is common practice to use canvas strips in making these connections. Motors and their mountings

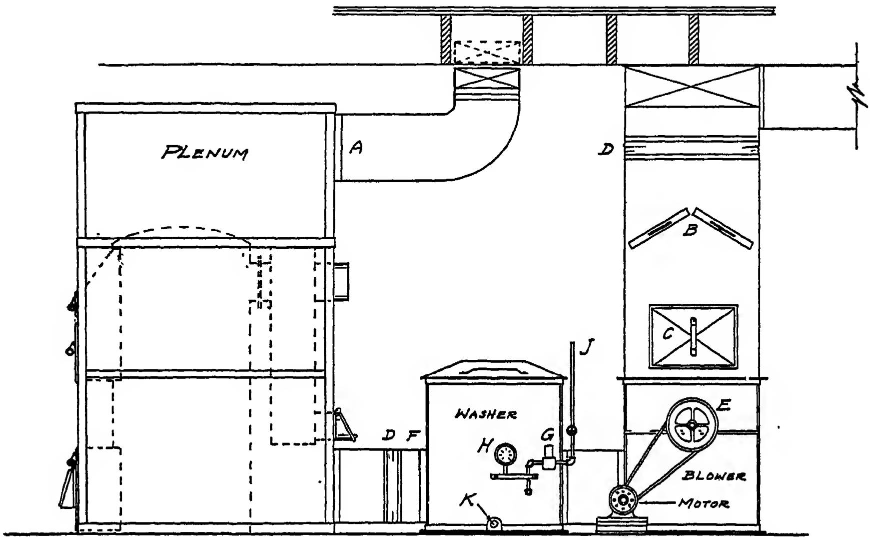

Fig. 3. Complete Fan Furnace Installation Showing Location of Fan, Furnace, Filters, Plenum Chamber and Method of Take-off of Warm Air Pipe

A. Transition fitting.
B. Filters.
C. Capped opening.
D. Canvas connection.
E. Pulley—3 diam. V-type.
F. Eliminator.
G. Solenoid valve.
H. Pressure gage.
J. Water supply.
K. Drain.

must be carefully chosen for quiet operation. Electrical conduit and water piping must not be fastened to, nor make contact with, fan housing. The installation of a fan directly under a cold air grille is not recommended on account of the noise objection. See also Chapter 18.

AIR WASHERS AND FILTERS

Washers may be provided in separate housings to be installed on the inlet or outlet side of the fan, or they may be integral with the fan construction. They operate at water pressures of from 10 to 30 lb and use two or more spray nozzles for washing and humidification. The sprays should be adjusted to completely cover the air passages.

Washers are usually controlled by solenoid valves wired in parallel with

the fan motor. The water supply may, in turn, be controlled by a humidity-controlling device located in one of the living rooms, so that the washer will operate at all times when the fan is in operation, unless the relative humidity should rise beyond a desirable percentage. Further information on washers will be found in Chapter 11.

There are many satisfactory types of filters on the market. These include dry filters, viscous filters, oil filters and other types, some of which must be cleaned, some of which must be cleaned and recharged with oil, and some of which are inexpensive and may be discarded when they become dirty, and replaced with new ones.

The resistance of a filter must be considered in the design of the system since the resistance rises rapidly as the filter becomes dirty, thus impairing the heating efficiency of the furnace, in fact, endangering the life of the furnace itself. Manufacturers' ratings of filters must be carefully regarded, and ample filter area must be provided. Filters must be replaced or cleaned when dirty. See also Chapter 16.

COOLING METHODS

Some cooling may be obtained under certain conditions by the use of basement air. A more positive cooling effect may be obtained through air washers where the temperature of the water is sufficiently low (55 F or lower), and where a sufficient volume of water can be provided. Unless the water is below the dew-point temperature of the indoor air at the time the washer is started, both the relative and absolute humidities will be somewhat increased.

Coils of copper finned tubing through which cold water is pumped are available for cooling. They require less space than air washers and have the advantage that no moisture is added to the air when the temperature of the water rises above the dew-point. Ample coil surface is necessary with this type of cooling.

It is thoroughly feasible to use ice or mechanical refrigeration in connection with the fan and duct system for the heating installation, and to cool the building by this method, provided the building is reasonably well constructed and insulated. Windows and doors should be tight, and awnings should be supplied on the sunny side of the building. See also Chapters 9 and 10.

Study of these problems sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in cooperation with the *National Warm Air Heating Association* is in progress at the University of Illinois. The following conclusions may be drawn from the studies thus far completed, subject to the limitations of the conditions under which the tests were run[1]:

1. An uninsulated building of ordinary residential type may require the equivalent of three tons of ice in 24 hours on days when the maximum outdoor temperature reaches 100 F if an effective temperature of approximately 72 deg is maintained indoors.

2. The use of awnings at all windows in east, south, and west exposures may result in savings of from 20 to 30 per cent in the required cooling load.

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[1]See A.S.H.V.E. research paper entitled, Study of Summer Cooling in the Research Residence at the University of Illinois, by A. P. Kratz and S. Konzo (A. S. H. V. E. Journal Section *Heating, Piping and Air Conditioning*, February, 1933).

3. The cooling load per degree difference in temperature is not constant but increases as the outdoor temperature increases.

4. The heat lag of the building complicates the estimation of the cooling load under any specified conditions and makes such estimates, based on the usual methods of computation, of doubtful value.

5. The seasonal cooling requirements are extremely variable from year to year, and the ratio between the degree-hours of any two seasons occurring within a 10-year period may be as high as 7.5 to 1. Hence an average value of the degree-hours cooling per season is comparatively meaningless.

6. The results of the tests suggest the use of a fan at night either to provide more comfortable conditions during the following day without provision for cooling, or to reduce the load required for cooling during the following day.

DUCT DESIGN

The ducts may be either round or rectangular. Rectangular ducts should be as nearly square as possible; the width should not be greater than four times the breadth. The radii of elbows should be not less than

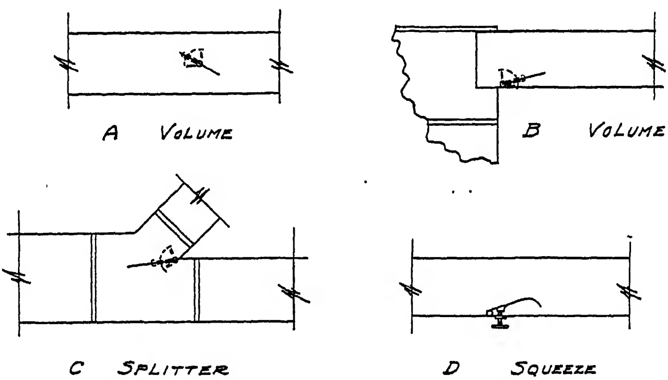

FIG. 4. THREE TYPES OF DAMPERS COMMONLY USED FOR TRUNK AND INDIVIDUAL DUCT SYSTEMS

one and one-half times the pipe diameter for round pipes, or the equivalent round pipe size in the case of rectangular ducts.

The ducts or piping may be designed either as a trunk line system or as a system of individual ducts from the furnace to each register. The engineering problems incident to the design of a trunk line system are somewhat more difficult than for the individual duct system. The trunk line system is generally a tailor-made job, whereas the individual duct system with which either round or square ducts may be used may frequently be assembled from stock materials and thus installed at a considerable saving. Individual ducts may frequently be grouped to simulate a trunk duct system in appearance. The design of ducts for air flow is described in Chapter 19.

Dampers

Suitable dampers are essential to any trunk or individual duct system, as it is virtually impossible to so lay out a system that it will be absolutely

in balance without the use of dampers. Special care must be used in the design of any system to avoid turbulence and to minimize resistance. Sharp elbows, angles, and offsets should be avoided. See Figs. 1 and 2, Chapter 19.

Three types of dampers are commonly used in trunk and individual duct systems. *Volume dampers* are used to completely cut off or reduce the flow through pipes. (See *A* and *B*, Fig. 4). *Splitter dampers* are used where a branch is taken off from a main trunk. (See *C*, Fig. 4). *Squeeze dampers* are used for adjusting the volume of air flow and resistance through a given duct. (See *D*, Fig. 4). It is essential that a damper be provided for each main or duct branch. A positive locking device should be used with each type of damper.

Supply and Return Air Registers

Supply registers may be located either in the floor, in a side wall at the baseboard, in a side wall at some point higher than the baseboard, or in the ceiling.

Velocities through registers may be reduced by the use of registers

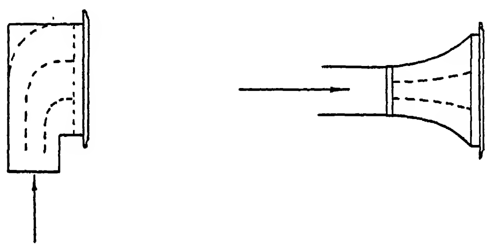

FIG. 5. DIFFUSERS IN TRANSITION FITTINGS TO EQUALIZE VELOCITIES THROUGH REGISTER FACES

larger than the connecting pipes. Some suggestions for equalizing velocities over the face area of the register by means of diffusers are illustrated in Fig. 5. Merely to use a larger register may not result in materially reduced velocities unless such diffusers are used.

Care should be exercised in making the connection between the supply register and its box to prevent streaking of the wall. All warm air registers should be equipped with dampers, or better, with diffuser dampers, which may be used to direct air currents in such a way that they will not be objectionable. (See Chapter 20).

CONTROLS

Air stratification, high bonnet temperatures, excessive flue gas temperatures, and heat overrun or lag in the system can be largely eliminated through proper care in the planning and installation of the control system. The essential requirements of the control are:

1. To keep the fire burning regardless of the weather.
2. To avoid excessive bonnet temperatures with resultant radiant heat losses into the basement.

3. To avoid the overheating of certain rooms through gravity action during off periods.
4. To have a sufficient supply of heat available at all times to avoid lag when the room thermostat calls for heat.
5. To avoid heat loss through the chimney by keeping stack temperatures low.
6. To provide quick response to the thermostat, with protection against overrun.
7. To provide for humidity control.
8. To provide a means of summer control of cooling.
9. To protect against fire hazards.

The following controls are desirable:

1. A *thermostat* located at a point where maximum fluctuation in temperature can be expected, in order to secure frequent operation of fans, drafts, and burners. This location would be near an outside wall, in a sun room, or in a room with some unusual exposure. The thermostat, of course, should not be located where it will be affected by direct radiant heat from the sun or from a fireplace, or by direct heat from any warm air duct or register.
2. A *furnacestat* to open and close the drafts according to bonnet temperature. The operating range of a furnacestat is approximately between 125 and 175 F, depending to a large extent upon outside weather conditions. In operation it may be necessary to adjust the setting for particular installations and readjust it for extremes of weather. The furnacestat should be installed either in the bonnet or in a main duct just outside the bonnet.
3. A *stack limit control* to shut down the system independently of the thermostat if the stack temperature exceeds 1000 F and thus prevent overheating of the furnace, and fuel waste.
4. On oil and gas burner installations, a control is usually included which will shut down the system if the fire goes out or if there is a failure of the ignition system.
5. A *humidistat* to regulate the moisture supplied to the rooms.
6. A device to open drafts, regardless of thermostat settings, whenever the bonnet temperature indicates that the fire is dying.

SELECTING THE FURNACE

The following formula may be used to compute the grate area of the furnace, assuming a ratio of heating surface to grate area of 20 to 1:

$$G = \frac{H}{F \times C \times E} \tag{1}$$

where

- G = required grate area, square feet.
- H = total heat loss from building, Btu per hour.
- F = calorific value of coal, Btu per pound.
- C = combustion rate in pounds of fuel per square foot of grate per hour.
- E = furnace efficiency based on heat available at register faces.

In practice it is customary to use the following constants:

- F = 13,000 (For specific values, see Fig. 1, Chapter 27).
- C = 5 to 10 lb (Use 8 lb as maximum in residence work).
- E = 55 per cent to 65 per cent depending on fuel burned. Lower efficiency must be used with highly volatile solid fuel.

Where ratio of heating surface to grate area is less or greater than 20 to 1, deduct or add 2 per cent from or to rating of furnace for each unit decrease or increase in ratio, as the case may be. The foregoing procedure

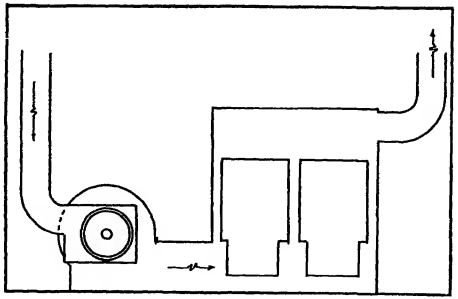

FIG. 6. HEATER ARRANGED FOR COMPLETE RECIRCULATION OF AIR WITH TEMPERATURE CONTROL DIRECT ON HEATER

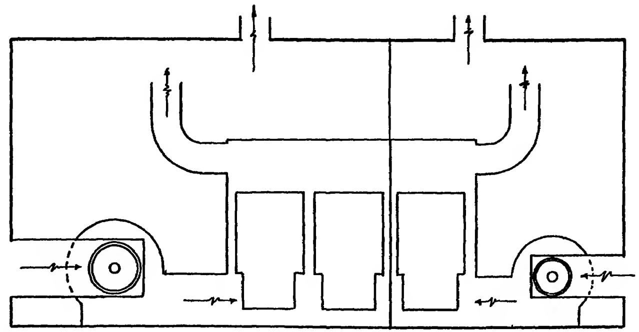

FIG. 7. TWO BATTERIES OF HEATERS AND FANS FOR INDEPENDENT SERVICE USING OUTSIDE AIR AND EXHAUST TO ATMOSPHERE

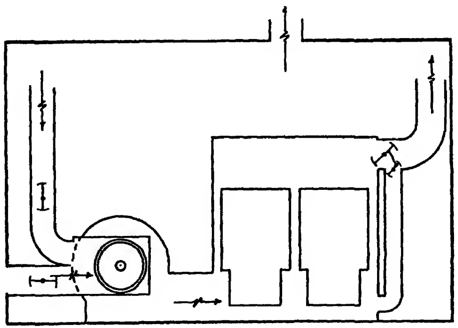

FIG. 8. HEATER ARRANGED FOR PARTIAL RECIRCULATION, ALSO SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR CHAMBERS, AND PARTIAL EXHAUST TO ATMOSPHERE

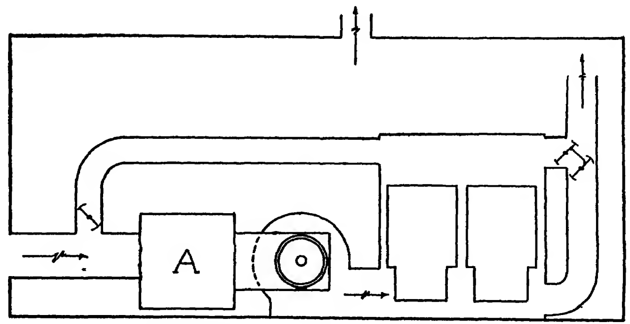

FIG. 9. HEATER ARRANGED FOR USE OF AIR WASHER OR FILTER (A) WITH HEATED AIR TO MIX WITH OUTSIDE AIR FOR TEMPERING, SHOWING MIXING DAMPER FROM WARM AIR AND TEMPERED AIR AND EXHAUST TO ATMOSPHERE

for determining the size of furnace to be used applies to continuously heated buildings. For intermittently heated buildings, it is advisable to increase the size by an amount ranging from 75 per cent to 200 per cent depending on the heat capacity of the construction materials, the higher percentage applying to materials of high heat capacity, such as concrete and brick.

Follow the same methods for an oil furnace as for coal where a conversion unit is to be used, making sure that the ratio of heating surface to grate area exceeds 20 to 1. If it does not, a size larger furnace should be selected. Use the manufacturers' Btu rating of furnaces designed for exclusive use with oil, and select a burner with liberal excess capacity.

Gas furnaces approved by the *American Gas Association*, which grants a Certificate of Approval to manufacturers, must carry the rated Btu per hour and official output in addition to an approval seal. Manufacturers' catalogs are also required to carry the output data.

The selection of the proper size gas furnace for a given installation can be easily made by using the following *A.G.A.* formula:

$$R = \frac{H}{0.9} \tag{2}$$

where

H = total heat loss from building in Btu per hour.
R = official *A.G.A.* output rating of the furnace in Btu per hour.

In the case of converted warm air furnaces a slightly different procedure is necessary, as the Btu input to the conversion burner must be selected rather than the furnace output. The proper sizing may be done by means of the following formula:

$$I = 1.56H \tag{3}$$

where

I = Btu per hour input.

The factor 1.56 is the multiplier necessary to care for a 10 per cent heat loss in the distributing ducts and an efficiency of 70 per cent in the conversion burner.

SELECTING THE FAN

Choose a fan which, according to its manufacturer's rating, is capable of delivering a volume of air, expressed in cubic feet per minute, against a frictional resistance, expressed in inches of water, computed by adding together the following items:

1. The frictional resistance of a warm air trunk or leader.
2. The frictional resistance of a return air trunk or duct.
3. The resistance to the flow of total volume of air through the furnace casing or hood, which is usually considered from 0.10 to 0.15 inches of water.
4. The frictional resistance through any other accessories, such as washers or filters.
5. A factor of safety of 10 per cent of the resistance calculated above.

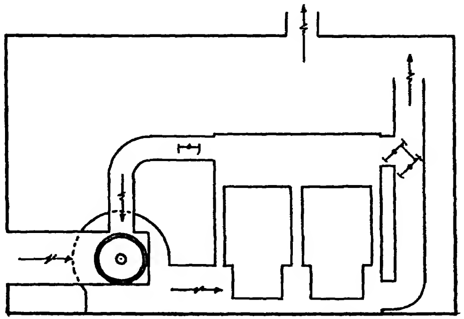

FIG. 10. HEATER ARRANGED TO USE COMPLETE OUTSIDE AIR WITH TEMPERED AIR CHAMBER ABOVE WARM AIR CHAMBER SHOWING MIXING DAMPER TO TEMPER AIR, ALSO MIXING DAMPER TO DUCT FROM WARM AND TEMPERED AIR CHAMBERS

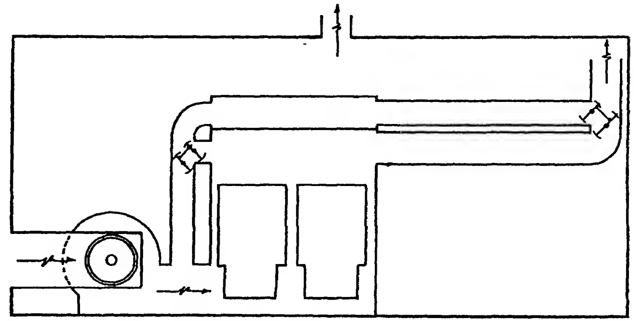

FIG. 11. HEATER ARRANGED TO USE OUTSIDE AIR, WITH TEMPERED AIR CHAMBER ABOVE WARM AIR CHAMBER SHOWING DOUBLE DUCT DISTRIBUTING SYSTEM AND MIXING DAMPERS AT BASE OF RISERS

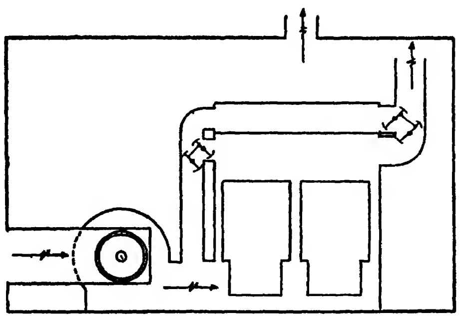

FIG. 12. HEATER ARRANGED FOR OUTSIDE AIR WITH WARM AIR CONNECTION TO FAN INLET TO TEMPER AIR, ALSO MIXING DAMPERS FROM WARM AIR AND TEMPERED AIR CHAMBER

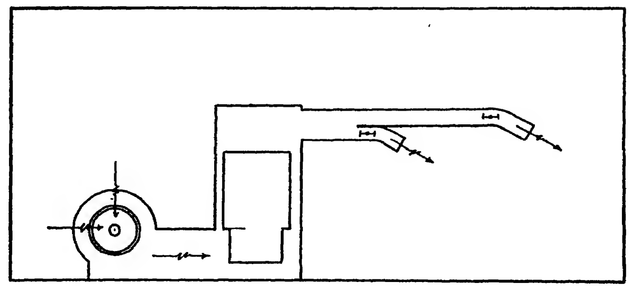

FIG. 13. HEATER ARRANGED FOR RECIRCULATION WITH TRUNK LINE DISTRIBUTING SYSTEM SHOWING TEMPERATURE CONTROL BY AIR VOLUME AT OUTLET, OR DIRECT ON HEATER

PROVISION FOR COOLING SYSTEM

If the system is to be used for cooling, the following provisions should be made:

1. Where cooling is to be secured through air circulation only:
   a. Provide for an increase of 25 to 50 per cent in fan capacity through multi-speed pulleys or other means.
   b. If basement air or outside night air is to be used, provide suitable basement opening in duct system, or outdoor air intake.
2. Where water *below* 55 F or artificial refrigeration or ice is to be used:
   a. Provide outside air duct for circulation of cool night air for economy.
   b. Make provision in return duct system for cooling unit.
   c. Make provision for reduction of fan speed.

HEAVY DUTY FAN FURNACES

Fan furnaces for large commercial and industrial buildings are available in sizes ranging from 400,000 to 1,800,000 Btu per hour per unit. Heavy duty heaters may be arranged in combinations of one or more units in a battery. A few possible arrangements are shown in Figs. 6 to 13, inclusive.

Control of temperature is secured through (1) controlling the quantity of heated air entering room, (2) using mixing dampers, or (3) regulating the fuel supply.

The design of heavy duty fan furnace heating systems is in many respects similar to that of the central fan heating systems described in Chapter 22. Ducts are designed by the method outlined in Chapter 19.

Chapter 24

GRAVITY WARM AIR SYSTEMS

Procedure for Design, Estimating Heating Requirements, Sizes of Leader Pipes, Proportioning Wall Stacks, Register Sizes, Recirculating Ducts and Grilles, Return Connection to Furnace, Furnace Capacity, Examples, Booster Fans

WARM air heating systems of the gravity type are described in this chapter[1], and those of the mechanical type are described in Chapter 23. In the gravity type, the motive head producing flow depends upon the difference in weight between the heated air leaving the top of the casing and the cooled air entering the bottom of the casing while in the mechanical type, a fan may supply all or part of the motive head. Booster fans are often used in conjunction with gravity-designed systems to increase air circulation.

In general, a warm-air furnace heating plant consists of a fuel-burning furnace or heater enclosed in a casing of sheet metal or brick, which is placed in the basement of the building. The heated air, taken from the top or sides near the top of the furnace casing, is distributed to the various rooms of the building through sheet metal warm-air pipes. The warm-air pipes in the basement are known as leaders, and the vertical warm-air pipes which are run in the inside partitions of the building are called stacks. The heated air is finally discharged into the rooms through registers which are set in register boxes placed either in the floor or in the side wall, usually at or near the baseboard.

The air supply to the furnace may be taken (1) entirely from inside the building through one or more recirculating ducts, (2) entirely from outside the building, in which case no air is recirculated, or (3) through a combination of the inside and the outside air supply systems.

PROCEDURE FOR DESIGN

The design of a furnace heating system involves the determination of the following items:

1. Heat loss in Btu from each room in the building.
2. Area and diameter in inches of warm-air pipes in basement (known as leaders).
3. Area and dimensions in inches of vertical pipes (known as wall stacks).
4. Free and gross area and dimensions in inches of warm-air registers.
5. Area and dimensions of recirculating or outside air ducts, in inches.
6. Free and gross area and dimensions in inches of recirculating registers.

[1]All figures and much of the engineering data which follow are from *Bulletins Nos.* 141, 188 and 189, Warm Air Furnaces and Heating Systems, Part II, by Professor A. C. Willard, A. P. Kratz, and V. S. Day, Engineering Experiment Station, University of Illinois.

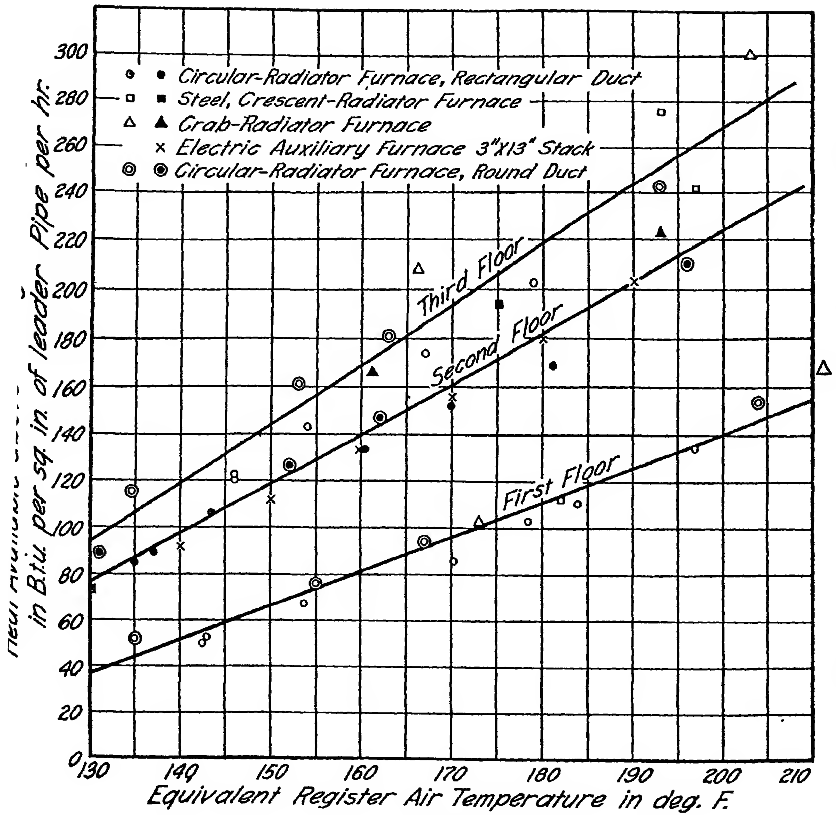

Fig. 1. Value of Square Inch of Leader Pipe Area for First, Second, and Third Floors for Simple System Having Leaders 8 Ft in Length

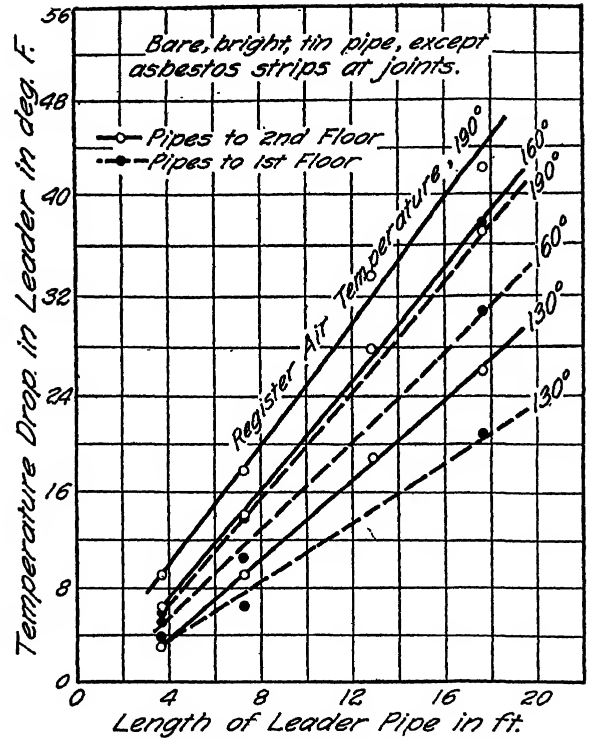

Fig. 2. Influence of Leader Pipe Length on Temperature Loss of Air Flowing through Pipe

7. Size of furnace necessary to supply the warm air required to overcome the heat loss from the building. This *size* should include square inches of leader pipe area which furnace must supply. It is also desirable to call for a minimum bottom fire-pot diameter in inches, which is the nominal grate diameter.

8. Area and dimensions in inches of chimney and smoke pipe. If an unlined chimney is to be used, that fact should be made clear.

The heat loss calculations should be made in accordance with the procedure outlined in Chapter 7, taking into consideration the transmission losses as well as the infiltration losses.

SIZES OF LEADER PIPES

In a gravity circulating warm-air furnace system the size of the leader to a given room depends upon the temperature of the warm air entering the room at the register. A reasonable air temperature at the registers must, therefore, be chosen before the system can be designed. The *National Warm Air Heating Association* has approved an air temperature of 175 F at the registers as satisfactory for design purposes. At this temperature, the heat-carrying capacity (heat available above 70 F) per square inch of leader pipe per hour for first, second or third floors is shown by Fig. 1 at 175 F to be 105, 170 and 208 Btu respectively. For average calculations, the values 111, 166 and 200 will simplify the work and may be satisfactorily substituted for these heat-carrying capacities. If H represents the total heat to be supplied any room, the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{111} = \text{approximately } 0.009H \tag{1}$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \tag{2}$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{200} = \text{approximately } 0.005H \tag{3}$$

In designing for a lower warm-air register temperature, say 160 F, the factors 111, 166 and 200 become 80, 140 and 166 (Fig. 1 at 160 F), and the resulting equations are:

$$\text{Leader areas for first floor, square inches} = \frac{H}{80} = \text{approximately } 0.012H \tag{4}$$

$$\text{Leader areas for second floor, square inches} = \frac{H}{140} = \text{approximately } 0.007H \tag{5}$$

$$\text{Leader areas for third floor, square inches} = \frac{H}{166} = \text{approximately } 0.006H \tag{6}$$

These equations are applicable to straight leaders from 6 to 8 ft in length. Longer leaders must be very thoroughly covered or else the vertical stacks must be increased in area as discussed under wall stacks. If some provision is not made for these longer leaders, the air temperature may be much lower than anticipated and the room will not be properly heated.

While Fig. 1 takes care of the drop in temperature in straight leaders up to 8 ft in length connected to stacks having about 75 per cent the

area of the leader, the designer must make allowances for all other conditions. The temperature drop in leaders of various lengths at three different register temperatures is shown in Fig. 2, and should be used to obtain new register temperatures, lower than 175 F, on which to base selections from the curves of Fig. 1, and thereby new constants for Equations 1, 2 and 3.

Leader sizes should in general be not less than obtained by Equations 1 to 3 nor should leaders less than 8 in. in diameter be used. It is not considered good commercial practice to specify diameters except

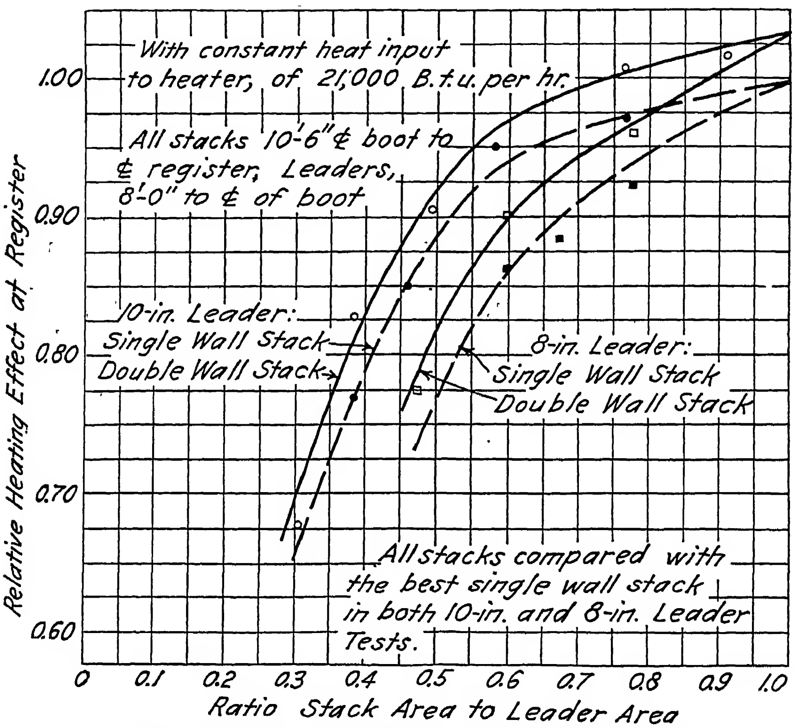

Fig. 3. Relative Heating Effect of Stacks at Constant Heat Input to Furnace

Note.—Pipe bare, bright tin except asbestos strips for joints.

in whole inches. The tops of leaders should be at the same elevation as they leave the furnace bonnet, and from this point there should be a uniform up-grade of 1 in. per foot of run in all cases. Leaders over 12 ft in length are to be avoided or should receive very special attention.

PROPORTIONING WALL STACKS

The wall stack for an upper floor should be made not less than 70 per cent of the area of the leader which has been selected from Fig. 1. So long as the leader is short and straight as was the case for Fig. 1, such a practice is probably justified, since the loss (Fig. 3) in capacity

occasioned by the smaller stack is not very serious for stacks having areas in excess of 70 per cent of the leader area. For leaders over 8 ft in length or for leaders which are not straight, the ratio of stack area to leader area should be greater than 70 per cent in order to offset the greater temperature losses (Fig. 2) in the longer leader. In gravity circulating systems, this stack to leader area ratio is a very important consideration. Specific data for a great variety of cases are presented in Figs. 4 and 5 and the designer should check the stack to leader combinations with the nearest comparable case as shown in these figures. Any second-floor stack supplying heat to a room whose heat loss is 9,000 Btu or more (see Figs. 4 and 5 which show that high temperatures are necessary if rooms of more than 9,000 Btu requirement are heated by one stack each in 4-in. studding), should be run within 6-in. studded walls or should have multiple stacks. Stack sections, wherever possible, should be changed from the thin rectangular to the more nearly square shape.

REGISTER SIZES

The registers used for discharging warm air into the rooms should have free or net area not less than the area of the leader in the same run of piping. The free area should be at least 70 per cent of the gross area of the register. No upper-floor register should be wider horizontally than the wall stack, and it should be placed either in the baseboard or side wall, if this can be done without the use of offsets. First-floor registers may be of the baseboard or floor type, with the former location preferred.

RECIRCULATING DUCTS AND GRILLES

The ducts through which air is returned to the furnace should be designed to minimize friction and turbulence. They should be of ample area, in excess of the total area of warm-air pipes, and at all points where the air stream must change direction or shape, streamline fittings should be employed. Horizontal ducts should pitch at least ½ in. per foot upward from the furnace.

The recirculating grilles (or registers) should have a free area at least equal to the ducts to which they connect, and their free area should never be less than 50 per cent of their gross area.

The location and number of return grilles will depend on the size, details and exposure of the house. Small compactly built houses may frequently be adequately served by a single return effectively placed in a central hall. More often it is desirable to provide two or more returns, provided, however, that in two-story residences one return must be placed to effectively receive the cold air returning by way of the stairs.

Where a divided system of two or more returns is used, the grilles must be placed to serve the maximum area of cold wall or windows. Thus in rooms having only small windows the grille should be brought as close to the furnace as possible, but if the room has a bay window, French doors, or other large sources of cooling or leakage of cold air, the grille should be placed close by, so as to collect the cool air and prevent drafts. When long ducts of this type are employed they must be made

oversize and favored in every way. This precaution is particularly important when long ducts and short ducts are used in the same system.

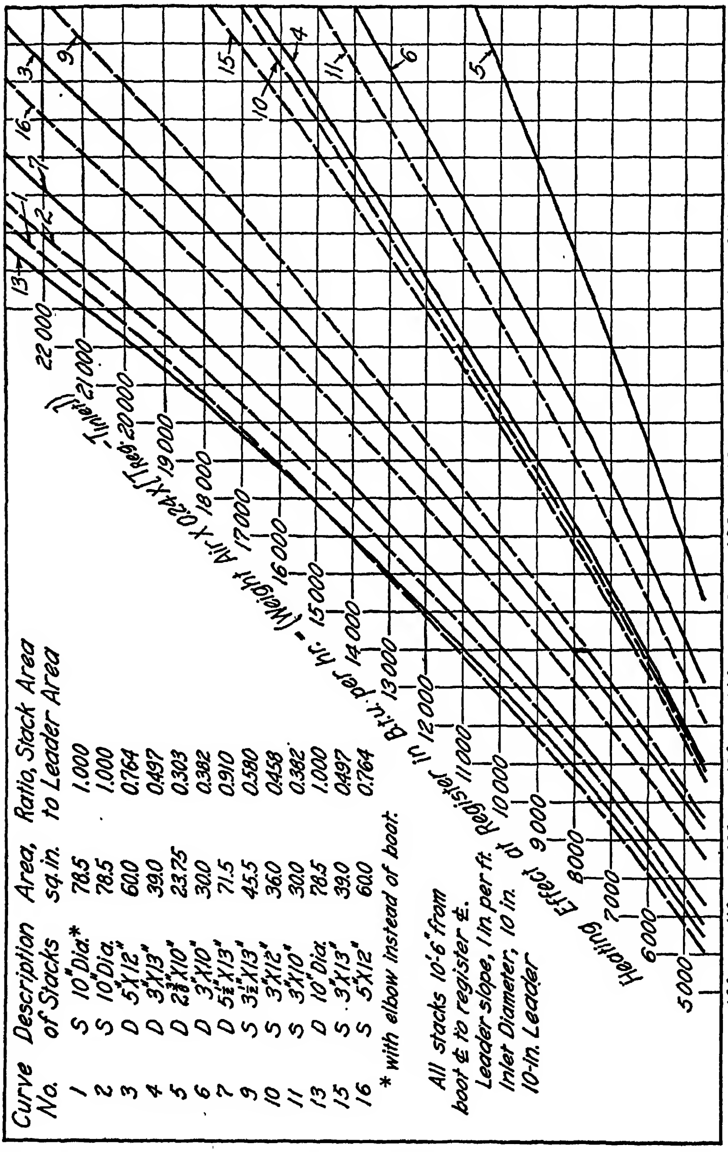

FIG. 4. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 10-IN. LEADER

The long ducts must be oversize, if they are to operate satisfactorily in parallel with short ducts.

Return ducts from upstairs rooms may be necessary in apartments or other spaces closed off or badly exposed. Metal linings are advisable in such ducts. It is important that these ducts be free from unnecessary

friction and turbulence, and that they be located to prevent preheating of the air before it reaches the furnace.

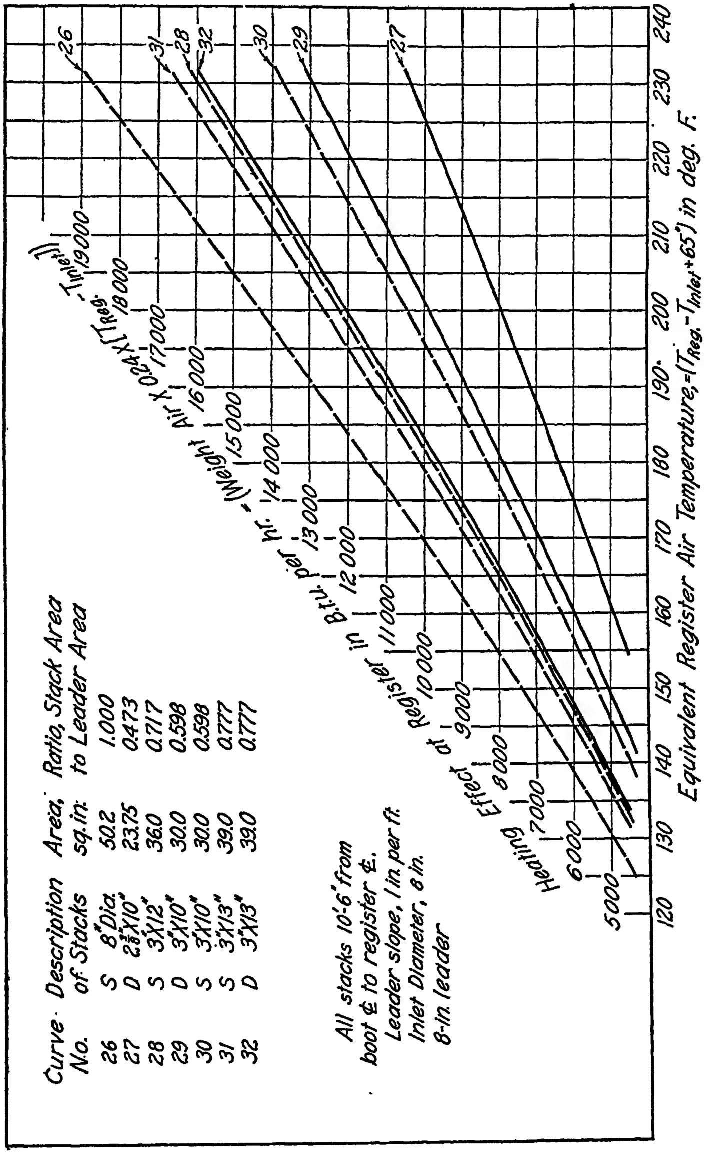

FIG. 5. HEATING EFFECT AT REGISTERS FOR VARIOUS STACKS WITH 8-IN. LEADER

Return Connection to Furnace

Circulation is accelerated if the drop to the furnace is through a round inclined pipe with, say, two 45-deg elbows rather than through a vertical drop and two 90-deg elbows. The top of the shoe should never enter

the casing above the level of the grate in the furnace. To accomplish this the shoe must be wide.

Tests of six different systems of cold air returns, Fig. 6, made at the University of Illinois[2], resulted in the following conclusions:

1. In general, somewhat better room temperature conditions may be obtained by returning the air from positions near the cold walls.

2. Friction and turbulence in elaborate return duct systems retard the flow of air, and may seriously reduce furnace efficiency, and lessen the advantages of such a design.

3. The cross-sectional duct-area is not the only measure of effectiveness. Friction and turbulence may operate to make the air flow out of all proportion to the various duct areas.

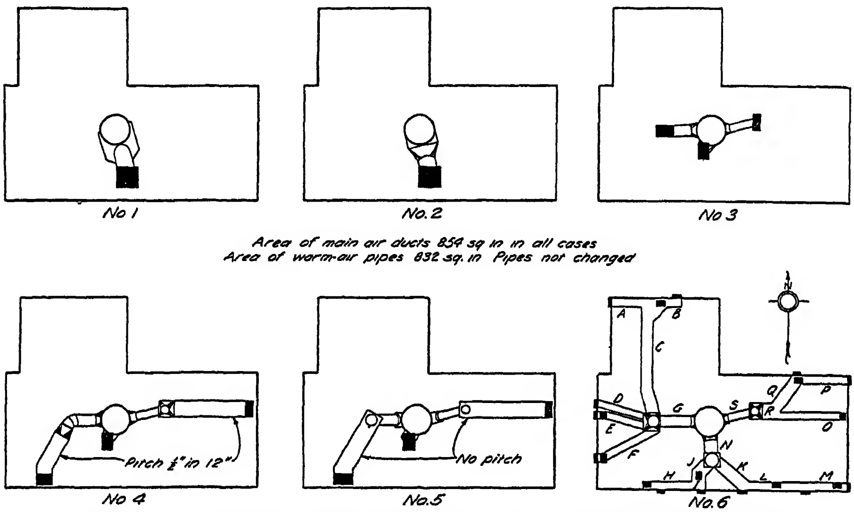

FIG. 6. ARRANGEMENT OF COLD AIR RETURNS FOR THE SIX INSTALLATIONS

FURNACE CAPACITY

The size of furnace should, of course, be such as will provide the necessary air heating capacity, usually expressed in square inches of leader pipe area, and at the same time provide a grate of the proper area to burn the necessary fuel at a reasonable chimney draft. The total leader pipe area required is easily obtained by finding the sum of the leader pipe areas as already designated.

The grate area will depend on several factors of which four are very important. First of all, the air temperature at the register for which the plant has been designed must be determined. Usually, this temperature is taken as 175 F. Second in importance is the combustion rate, *which must always correspond with the register air temperature*, as is shown by reference to a set of typical furnace performance curves (Fig. 7) for a cast-iron circular radiator furnace with a 23-in. diameter grate and 50-in. diameter casing. The conditions shown on these curves which seem to

[2]Investigation of Warm-Air Furnaces and Heating Systems, Part IV, by A. C. Willard, A. P. Kratz and V. S. Day (University of Illinois *Engineering Experiment Station Bulletin No.* 189).

approximate nearest to the 175 F register warm-air temperature are—combustion rate, 7 lb; warm-air register temperature, 173 F; efficiency of the furnace, 58.5 per cent. The third factor is efficiency, which, in turn, is a function of the combustion rate varying with it as shown by the efficiency curve of Fig. 7. The fourth factor is the heat value per pound of fuel burned, which was 12,790 Btu, but is not shown on the curves since it was constant for all combustion rates.

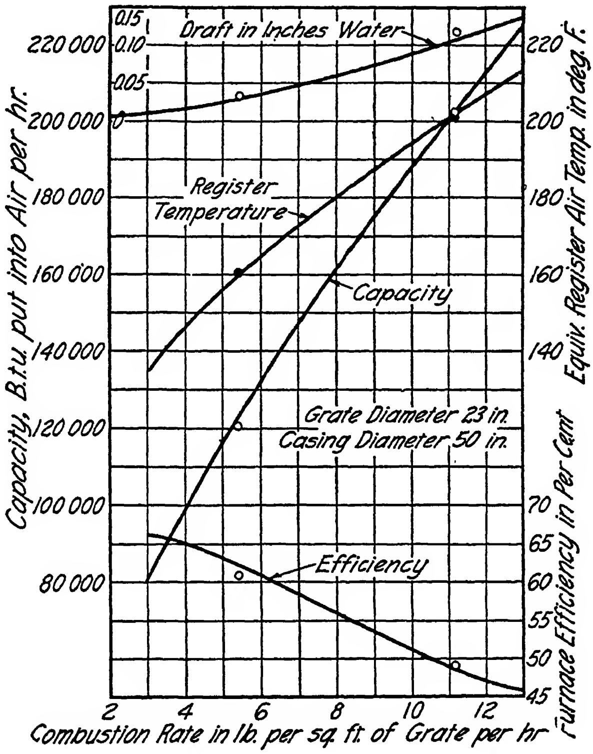

FIG. 7. TYPICAL PERFORMANCE CURVES FOR A WARM AIR FURNACE AND INSTALLATION IN A THREE-STORY TEN LEADER PLANT, OPERATING ON RECIRCULATED AIR

From the relation existing between these factors it is found (Fig. 7) that the capacity of the furnace under test is 147,750 Btu per hour for the total grate, which gives the capacity at the furnace bonnet per square foot of grate as 51,200 Btu and per square inch of grate as 356 Btu per hour.

Suppose it is desired to select a furnace to deliver air to the rooms at a register temperature approximating 160 F rather than 175 F. Referring to the curves, the relation is—combustion rate, 5.5 lb; register warm-air temperature, 160 F; and efficiency of the furnace, 62 per cent. Under this condition the capacity of the furnace at the furnace bonnet per square foot of grate area is 43,200 Btu per hour, and per square inch of grate it is 300 Btu per hour. From these performance values, the grate area for any

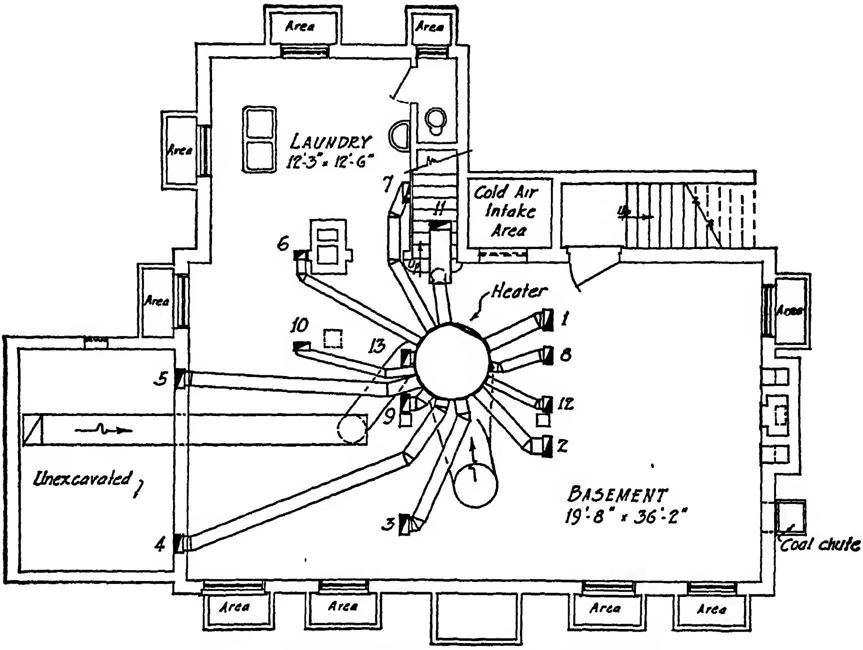

FIG. 8. BASEMENT PLAN, RESEARCH RESIDENCE

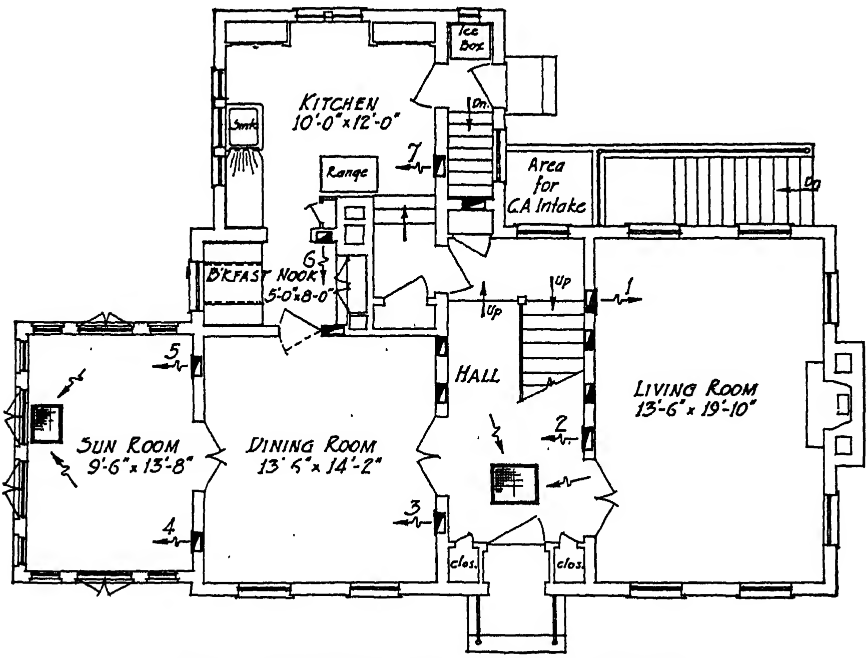

FIG. 9. FIRST-FLOOR PLAN, RESEARCH RESIDENCE

plant requirement will be (allowing 20 per cent heat loss between furnace and registers):

$$\text{Grate area (175 F register temperature), square inches} = \frac{1.2\ H}{356} = 0.0034H^{*} \quad (7)$$

$$\text{Grate area (160 F), square inches} = \frac{1.2\ H}{300} = 0.0040H^{*} \quad (8)$$

It is not always possible to obtain performance curves, and the following method is suggested as being a close check. An addition of 2 per cent of the furnace capacity is proposed for each unit that the heating surface to grate area ratio of the furnace exceeds 20. This addition is based on tests of four types of furnaces having various ratios of heating surface to grate area, at the University of Illinois.

Let E = efficiency of the furnace.
f = fuel value of the coal, Btu per pound.
p = pounds of coal burned per square foot grate per hour.
R = ratio of heating surface to grate area.
H = total heat requirements of the house.

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{Efp\ [1 + 0.02\ (R - 20)]} \text{ for all inside air} \quad (9)$$

For coal having a heat value of 12,000 Btu, and a furnace having 60 per cent efficiency, with 6 lb of coal burned per square foot of grate per hour, and 20 sq ft of heating surface for 1 sq ft of grate, this becomes:

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{0.60 \times 12{,}000 \times 6} \text{ for all inside air} \quad (10)$$

and for another furnace having 24 sq ft of heating surface for 1 sq ft of grate the expression is

$$\text{Grate area, square inches} = \frac{1.2 \times 144\ H}{0.60 \times 12{,}000 \times 6\ [1 + 0.02\ (24 - 20)]} \quad (11)$$

The air temperatures at the registers corresponding to the conditions of Equation 11 would be approximately 165 F, and for 175 F and 12,000 Btu the combustion rate would be about 7.5 lb with an efficiency of 57 per cent, using the curves of Fig. 7 as a guide.

EXAMPLES

The application of the preceding data to an actual example may be of assistance to the designer. Figs. 8, 9, 10 and 11 represent the plans of the Warm Air Research Residence of the *National Warm Air Heating Association* erected at the University of Illinois[3].

[3]Plans used with permission. Bathroom on third floor not heated.

*Let H = Btu heat loss from the entire house per hour = summation of all room losses $H_1 + H_2$ + etc. + the Btu necessary to heat the fresh air, if any, at intake. This fresh air loss in Btu per hour will be approximately 1.27 times the cubic feet of air admitted through the intake per hour on a zero day. For systems which recirculate all the air this value will be zero. For systems which have a fresh air intake, controlled by damper, this value might well be approximated, since this loss will probably be reduced to a minimum on a zero day. Assume for such cases, that the building loss is increased by 25 per cent, and that there is the usual 20 per cent loss between furnace and registers.

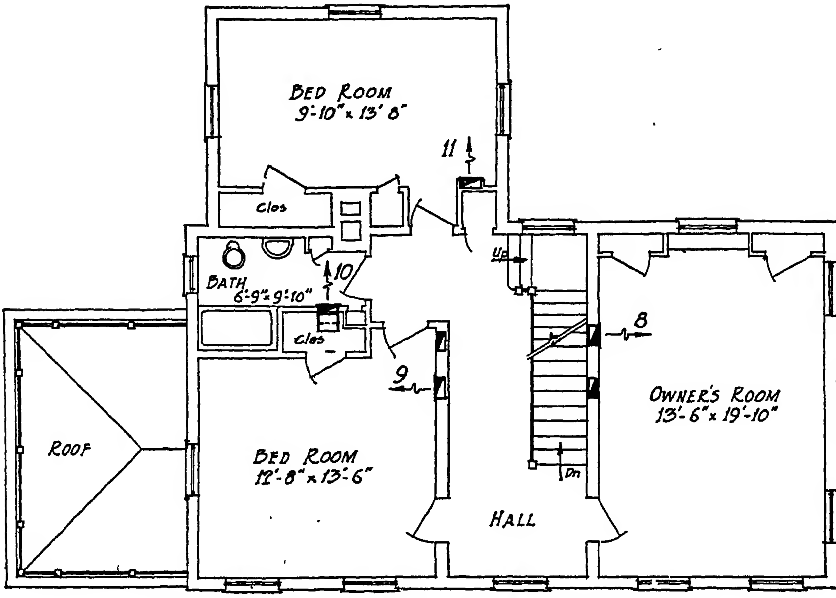

Fig. 10. Second-Floor Plan, Research Residence

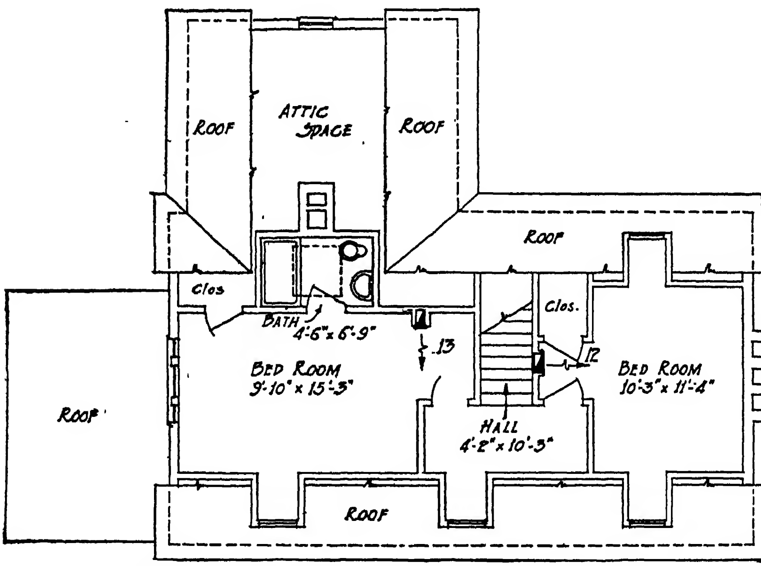

Fig. 11. Third-Floor Plan, Research Residence

Leaders, Stacks and Registers. (Direct Method)

Living Room, 1st floor:

17,250 ÷ 111 = 155 sq in. leader area. See summary, Table 1; also example under Standard Code[4], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 14 in.

Register size = 155 sq in. net area. Gross area = net area ÷ 0.7 = 14 × 16 in.

Owner's Room, 2nd floor:

15,030 ÷ 167 = 90 sq in. leader area. See Summary Table; also example under Standard Code[4], Art. 3, Basis of Working Rules for Pipes.

Leader diameter = 11.4, say 12 in.

Stack area = 0.7 × 90 = 63 sq in. = say 5 × 12 in.

Register area = 90 sq in. net area. Gross area = net area ÷ 0.7 = 12 × 12 or 12 × 14 in.

In like manner the leaders, stacks and registers are calculated for each room in the house.

Leaders, Stacks and Registers. (Code[4] Method. See Art. 3, Sec. 1, 2, 3.)

Living Room (Glass = 90, Net wall = 405, Cubic contents = 2405)

$$\text{Leader} = \left(\frac{90}{12} + \frac{405}{60} + \frac{2405}{800}\right) 9 = 155 \text{ sq in.}$$

Register, same as Direct Method.

Owner's Room (Glass = 68, Net wall = 394, Cubic contents = 2275)

$$\text{Leader} = \left(\frac{68}{12} + \frac{394}{60} + \frac{2275}{800}\right) 6 = 90 \text{ sq in.}$$

Stack and Register, same as Direct Method.

Assuming all air recirculated, the minimum furnace for the plant will be:

Grate Area = 0.0034 × 132,370 = 450 sq in. = 24 in. diam. at 175 F. register temperature. (Equation 7)

Grate Area = 0.0040 × 132,370 = 530 sq in. = 26 in. diam. at 160 F. register temperature. (Equation 8)

If provision shall be made for certain outside air circulation, then increase the building heat loss by, say 25 per cent and obtain by Equation 7 a 27-in. grate and by Equations 8 and 10 a 29-in. grate.

Experiments at the University of Illinois[5] have shown that the capacity of a furnace may be increased nearly three times by an adequate fan, with a constant register or delivery temperature maintained, *provided that the rate of fuel consumption can be increased to provide the necessary heat.* In other words, the capacity of a forced circulation system *is limited by the ability of the chimney to produce a sufficient draft.*

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[4]Standard Code Regulating the Installation of Gravity Warm Air Heating Systems in Residences. This code has been sponsored by the *National Warm Air Heating Association*, the *National Association of Sheet Metal Contractors*, and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS. It is recommended that the installation of all gravity warm air heating systems in residences be governed by the provisions of this code, the eighth edition of which may be obtained from the *National Warm Air Heating Association*, 3440 A.I.U. Building, Columbus, Ohio.

[5]See University of Illinois *Eng. Exp. Sta. Bulletin No.* 120, p. 129.

TABLE 1. SUMMARY OF DATA APPLIED TO WARM AIR RESEARCH RESIDENCE

Rooms	From Chapter 7 Estimating Heat Losses Btu Heat Losses H	Leader Area Sq In.	Stack Area Sq In. $0.7 \times LA$	Leader Diameter Inches	Stack Size Net	Register Size Gross
First Floor		$= 0.009H$				
Living	17250	155	---	14	....	14×16
Dining	6810	61	---	9	....	8×12
Breakfast	2300	21	---	8	....	8×10
Kitchen	9210	83	---	11 or 12	....	12×14
Sun	25710	230	---	Two 12	....	Two 12×14
Hall and stair	12570	113	---	12	....	12×14
Second Floor		$= 0.006H$				
Owners	15030	90	63	11 or 12	5×12	12×14
S. W. Bed	9800	59	41	9	$3\frac{1}{2} \times 12$	8×12
Bath	2450	15	10	8	3×10	8×10
N. Bed	14800	89	62	11 or 12	5×12	12×14
Third Floor		$= 0.005H$				
E. Bed	8220	41	29	8	3×10	8×10
W. Bed	8220	41	29	8	3×10	8×10

BOOSTER FANS

Booster fans often may be arranged to operate when the gas and oil burners operate and to stop automatically when the burners shut down. The booster equipment is most effective in increasing output at low operating temperatures. According to tests, efficiencies may be advanced from 60 per cent for gravity to 70 per cent with boosters at low operating temperatures, but at high operating temperatures gravity and booster efficiencies are almost identical[6].

[6]See University of Illinois *Eng. Exp. Sta. Bulletin No.* 141, p. 79.

Chapter 25

HEATING BOILERS

Cast-Iron Boilers, Steel Boilers, Special Heating Boilers, Hot Water Supply Boilers, Furnace Design, Heating Surface, Testing and Rating Codes, Output, Efficiency, Selection of Boilers, Connections and Fittings, Erection, Operation and Maintenance, Boiler Insulation

STEAM and hot water boilers for low pressure heating work are built in a wide variety of types, many of which are illustrated in the Catalog Data Section, and are classified as (1) cast-iron sectional, (2) steel fire tube, (3) steel water tube, and (4) special.

CAST-IRON BOILERS

Cast-iron boilers may be of round pattern with circular grate and horizontal pancake sections joined by push nipples and tie rods, or of rectangular pattern with vertical sections. The latter type may be either of outside header construction where each section is independent of the other and the water and steam connections are made externally through these headers, or assembled with push nipples and tie rods, in which case the water and steam connections are internal.

Cast-iron boilers usually are shipped knocked down to facilitate handling at the place of installation where assembly is made. One of the chief advantages of cast-iron boilers is that the separate sections can be taken into or out of basements and other places more or less inaccessible after the building is constructed. This feature is of importance in making repairs to or replacing a damaged or worn out boiler and should be given consideration in the original selection. Sufficient space should be provided in the boiler room for assembling the boiler and for disassembling it conveniently if repairs are needed. With the outside header type of boiler a damaged section in the middle of the boiler can be removed without disturbing the other sections and sufficient side clearance should be provided for this contingency.

Capacities of cast-iron boilers range from that required for small residences up to about 18,000 sq ft of steam radiation. For larger loads, cast-iron boilers must be installed in multiple or a steel boiler used. In most cases cast-iron boilers are limited to working pressures of 15 lb for steam and 30 lb for water. Special types are built for hot water supply which will withstand higher local water pressures.

STEEL BOILERS

Two general classifications may be applied to steel boilers: *first*, with regard to the relative position of water and hot gases, distinguished as fire

tube or water tube; *second*, with regard to arrangement of furnace and flues, as (1) horizontal return tubular (HRT) boilers, (2) portable (self-contained) firebox boilers with either water or fire tubes, and (3) water tube boilers of the power type.

Fire tube boilers are constructed so that the water available to produce steam is contained in comparatively large bodies distributed outside of the boiler tubes, the hot gases passing within the tubes. In *water tube* boilers, the water is circulated within the boiler tubes, heat being applied externally to them.

The *HRT boiler* is the oldest type and consists of a horizontal cylindrical shell with fire tubes, enclosed in brickwork to form the furnace and combustion chamber. All heating surfaces and the interior of the boiler are accessible for both cleaning and inspection. Horizontal return tubular boilers should be suspended from structural columns and beams independent of the brick setting, especially the larger sizes. Small HRT boilers sometimes are supported by brackets resting on the brick setting.

TABLE 1. PRACTICAL COMBUSTION RATES FOR SMALL COAL-FIRED HEATING BOILERS OPERATING ON NATURAL DRAFT OF FROM ¼ IN. TO ½ IN. WATER[a]

KIND OF COAL	SQ FT GRATE	LB OF COAL PER SQ FT GRATE PER HOUR
No. 1 Buckwheat Anthracite	Up to 4	3
	5 to 9	3½
	10 to 14	4
	15 to 19	4½
	20 to 25	5
Anthracite Pea	Up to 9	5
	10 to 19	5½
	20 to 25	6
Anthracite Nut and Larger	Up to 4	8
	5 to 9	9
	10 to 14	10
	15 to 19	11
	20 to 25	13
Bituminous	Up to 4	9.5
	5 to 14	12
	15 and above	15.5

[a]Steel boilers usually have higher combustion rates for grate areas exceeding 15 sq ft than those indicated in this table.

Portable firebox boilers are the more generally used type of steel heating boilers, their outstanding characteristic being the water-jacketed firebox which eliminates virtually all brickwork. They are shipped in one piece from the factory and come to the job ready for immediate hook-up to piping. They may be of welded or riveted construction and have either water or fire tubes. Manufacturers' catalogs usually list heating surface as well as grate area. The elimination of brickwork also makes this type the most compact of steel boilers as well as the lowest in first cost.

Water tube boilers. For large heating loads water tube boilers are quite frequently used. They usually require more head room than other types of boilers but require considerably less floor space and make possible a

much higher rate of evaporation per square foot of heating surface, with proper setting, baffling and draft. Water tube boilers used for heating purposes are brick set, supported on structural steel columns and have the brick setting encased in an insulated steel housing to prevent air infiltration and to minimize heat losses. For large heating loads at a high rate of evaporation, such boilers should be operated at pressures above 15 lb per square inch with a pressure-reducing valve on the connection to the heating main.

SPECIAL HEATING BOILERS

A special type of boiler, known as the *magazine feed boiler*, has been developed for the burning of small sizes of anthracite. These are built of both cast-iron and steel, and have a large fuel carrying capacity which results in longer firing periods than would be the case with the standard types using buckwheat sizes of coal. Special attention must be given to insure adequate draft and proper chimney sizes and connections.

Oil-burner boiler units, in which a special boiler has been designed with a furnace shaped to suit the particular burner used, have been developed by a number of manufacturers. These usually are compact units with the burner and all controls enclosed within an insulated steel jacket. Ample furnace volume is provided for efficient combustion, and the heating surfaces are proportioned for effective heat transfer. Consequently, higher efficiencies are obtainable than with the ordinary coal fired boiler converted to oil firing.

GAS-FIRED BOILERS

Gas boilers have assumed a well-defined individuality. The usual boiler is sectional in construction with a number of independent burners placed beneath the sections. In most boilers each section has its own burner. In all cases the sections are placed quite closely together, much closer than would be possible when burning a soot-forming fuel. The effort of the designer is always to break the hot gas up into thin streams, so that all particles of the heat-carrying gases can come as closely as possible to the heat-absorbing surfaces. Because there is no fuel bed resistance and because the gas company supplies the motive power to draw in the air necessary for combustion (in the form of the initial gas pressure), draft losses through gas boilers are low.

HOT WATER SUPPLY BOILERS

Boilers for hot water supply are classified as direct, if the water heated passes through the boiler, and as indirect, if the water heated does not come in contact with the water or steam in the boiler.

Direct heaters are built to operate at the pressures found in city supply mains and are tested at pressures from 200 to 300 lb per square inch. The life of direct heaters depends almost entirely on the scale-making properties of the water supplied. If water temperatures are maintained below 140 F the life of the heater will be much longer than if higher temperatures are used, owing to decreased scale formation and minimized corrosion below 140 F. Direct water heaters in some cases are designed to burn refuse and garbage.

Indirect heaters generally consist of steam boilers in connection with heat exchangers of the coil or tube types which transmit the heat from the steam to the water. This type of installation has the following advantages:

1. The boiler operates at low pressure.
2. The boiler is protected from scale and corrosion.
3. The scale is formed in the heat exchanger in which the parts to which the scale is attached can be cleaned or replaced. The accumulation of scale does not affect efficiency although it will affect the capacity of the heat exchanger.
4. Discoloration of water may be prevented if the water supply comes in contact with only non-ferrous metal.

Where a steam heating system is installed, the domestic hot water usually is obtained from an indirect heater placed below the water line of the boiler.

FURNACE DESIGN

Good efficiency and proper boiler performance are dependent on correct furnace design embodying sufficient volume for burning the particular fuel at hand, which requires thorough mixing of air and gases at a high temperature with a velocity low enough to permit complete combustion of all the volatiles. On account of the small amount of volatiles contained in coke, anthracite and semi-bituminous coal, these fuels can be burned efficiently with less furnace volume than is required for bituminous coal, the combustion space being proportioned according to the amount of volatiles present.

Combustion should take place before the gases are cooled by the boiler heating surface, and the volume of the furnace must be sufficient for this purpose. The furnace temperature must be maintained sufficiently high to produce complete combustion, thus resulting in a higher CO_2 content and the absence of CO. Hydrocarbon gases ignite at temperatures varying from 1000 to 1500 F.

The question of furnace proportions, particularly in regard to mechanical stoker installations, has been given some consideration by various manufacturers' associations. Arbitrary values have been recommended for minimum dimensions. A customary rule-of-thumb method of figuring furnace volumes is to allow 1 cu ft of space for a maximum heat release of 50,000 Btu per hour. This value is equivalent to allowing approximately 1 cu ft for each developed horsepower, and it is approved by most smoke prevention organizations.

The setting height will vary with the type of stoker. In an overfeed stoker, for instance, all the volatiles must be burned in the combustion chamber and, therefore, a greater distance should be allowed than for an underfeed stoker where a considerable portion of the gas is burned while passing through the incandescent fuel bed. The design of boiler also may affect the setting height, since in certain types the gas enters the tubes immediately after leaving the combustion chamber while in others it passes over a bridge wall and toward the rear, thus giving a better opportunity for combustion by obtaining a longer travel before entering the tubes.

To secure suitable furnace volume, especially for mechanical stokers or oil burning, it often is necessary either to pit the stoker or oil burner, or

where water line conditions and headroom permit, to raise the boiler on a brick foundation setting.

Smokeless combustion of the more volatile bituminous coals is furthered by the use of mechanical stokers. (See Chapter 28). Smokeless combustion in hand-fired boilers burning high volatile solid fuel is aided (1) by the use of double grates with down-draft through the upper grate, (2) by the use of a curtain section through which preheated auxiliary air is introduced over the fire toward the rear of the boiler, and (3) by the introduction of preheated air through passages at the front of the boiler. All three methods depend largely on mixing secondary air with the partially burned volatiles and causing this mixture to pass over an incandescent fuel bed, thus tending to secure more complete combustion than is possible in boilers without such provision.

BOILER HEATING SURFACE

Boiler heating surface is that portion of the surface of the heat transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other side. (See definition in Chapter 42). Heating surface on which the fire shines is known as *direct* or radiant surface and that in contact with hot gases only, as *indirect* or convection surface. The amount of heating surface, its distribution and the temperatures on either side thereof influence the capacity of any boiler.

Direct heating surface is more valuable than indirect per square foot because it is subjected to a higher temperature and also, in the case of solid fuel, because it is in position to receive the full radiant energy of the fuel bed. The heat transfer capacity of radiant heating surface may be as high as 6 to 8 times that of indirect surface. This is one of the reasons why the water legs of some boilers have been extended, especially in the case of stoker firing where the extra amount of combustion chamber secured by an extension of the water legs is important. For the same reason, care should be exercised in building a refractory combustion chamber in an oil-burning boiler so as not to screen any more of this valuable surface with refractories than is necessary for good combustion.

The effectiveness of the heating surface depends on its cleanliness, its location in the boiler, and the shape of the gas passages. Investigations[1] by the U. S. Bureau of Mines show that:

1. A boiler in which the heating surface is arranged to give long gas passages of small cross-section will be more efficient than a boiler in which the gas passages are short and of larger cross-section.
2. The efficiency of a water tube boiler increases as the free area between individual tubes decreases and as the length of the gas pass increases.
3. By inserting baffles so that the heating surface is arranged in series with respect to the gas flow, the boiler efficiency will be increased.

The area of the gas passages must not be so small as to cause excessive resistance to the flow of gases where natural draft is employed.

Heat Transfer Rates

Practical rates of heat transfer in heating boilers will average about

[1]See *U. S. Bureau of Mines Bulletin No.* 18, The Transmission of Heat into Steam Boilers.

3300 Btu per sq ft per hour for hand-fired boilers and 4000 Btu per sq ft per hour for mechanically fired boilers when operating at *designed load*[2]. When operating at *maximum load*[2] these values will run between 5000 and 6000 Btu per sq ft per hour. Boilers operating under favorable conditions at the above heat transfer rates will give exit gas temperatures that are considered consistent with good practice.

TESTING AND RATING CODES

The Society has adopted three solid fuel testing codes, a solid fuel rating code and one oil fuel testing code. A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers—Codes 1 and 2—(Revision of June 1929)[3], are intended to provide a method for conducting and reporting tests to determine heat efficiency and performance characteristics. A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers—Code No. 3—(Edition of 1929)[3] is intended for use with A.S.H.V.E. Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers[4]. The object of this test code is to specify the tests to be conducted and to provide a method for conducting and reporting tests to determine the efficiencies and performance of the boiler. The A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel[5] is intended to provide a standard method for conducting and reporting tests to determine the heating efficiency and performance characteristics when oil fuel is used with steam heating boilers. The *Steel Heating Boiler Institute* suggests a single number dimensional rating in the *S.H.B.I.* Code for the Rating of Low-Pressure Heating Boilers by Their Physical Characteristics[6].

BOILER OUTPUT

Boiler output as defined in A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3) is the quantity of heat available at the boiler nozzle with the boiler normally insulated. It should be based on actual tests conducted in accordance with this code. This output is usually stated in Btu and in square feet of equivalent heating surface (radiation). According to the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers, the performance data should be given in tabular or curve form on the following items for at least five outputs ranging from maximum down to 35 per cent of maximum: (1) fuel available, (2) combustion rate, (3) efficiency, (4) draft tension, (5) flue gas temperature. The only definite restriction placed on setting the maximum output is that priming shall not exceed 2 per cent. These curves provide complete data regarding the performance of the boiler under test conditions. Certain other pertinent information, such as grate area, heating surface and chimney dimensions is desirable also in forming an opinion of how the boiler will perform in actual service.

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[2]For definition of design load and maximum load see pages 337 and 338.

[3]See A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929. Also Chapter 42.

[4]See A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930. Also Chapter 42.

[5]See A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931. Also Chapter 42.

[6]See Rating of Heating Boilers by Their Physical Characteristics, by C. E. Bronson (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930).

The output of large heating boilers is frequently stated in terms of *boiler horsepower* instead of in Btu per hour or square feet of equivalent radiation.

Boiler Horsepower: The evaporation of 34.5 lb of water per hour *from and at 212 F* which is equivalent to a heat output of $970.2 \times 34.5 =$ 33,471.9 Btu per hour.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at this same temperature and at atmospheric pressure.

It is usually considered that 10 sq ft of boiler heating surface will produce a rated boiler horsepower. A rated boiler horsepower in turn can carry a design load of from 100 to 140 sq ft of equivalent radiation. It is apparent, therefore, that 1 sq ft of boiler heating surface can carry a design load of from 10 to 14 sq ft of equivalent radiation, or somewhat more if the boiler is forced above rating. The application of these values is discussed under the heading Selection of Boilers.

BOILER EFFICIENCY

The term *efficiency* as used for guarantees of boiler performance is usually construed as follows:

1. *Solid Fuels.* The efficiency of the *boiler* alone is the ratio of the heat absorbed by the water and steam in the boiler per pound of combustible burned on the grate to the calorific value of 1 lb of combustible as fired. The *combined* efficiency of *boiler, furnace and grate* is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel as fired to the calorific value of 1 lb of fuel as fired.

2. *Liquid Fuels.* The *combined* efficiency of boiler, furnace and burner is the ratio of the heat absorbed by the water and steam in the boiler per pound of fuel to the calorific value of 1 lb of fuel.

Solid fuel boilers usually show an efficiency of 50 to 75 per cent when operated under favorable conditions at their rated capacities. Information on the combined efficiencies of boiler, furnace and burner has resulted from research conducted at Yale University in cooperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*[7]. For general information on heating efficiencies see Chapter 29.

SELECTION OF BOILERS

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated design load is the sum of the following three items[8]:

1. The estimated heat emission in Btu per hour of the connected radiation (direct, indirect or central fan) to be installed.

[7]See A.S.H.V.E. research papers entitled Study of the Characteristics of Oil Burners and Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931), and A Study of Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

2. The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler.

3. The estimated heat emission in Btu per hour of the piping connecting the radiation and other apparatus to the boiler.

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—Edition of April, 1932).

The estimated maximum load is given by[8]:

4. The estimated increase in the normal load in Btu per hour due to starting up cold radiation. This percentage of increase is to be based on the sum of Items 1, 2 and 3 and the heating-up factors given in Table 2.

TABLE 2. WARMING-UP ALLOWANCES FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS[a-b-c]

DESIGN LOAD (REPRESENTING SUMMATION OF ITEMS 1, 2, AND 3,[d]		PERCENTAGE CAPACITY TO ADD FOR WARMING UP
Btu per Hour	Equivalent Square Feet of Radiation[d]	
Up to 100,000	Up to 420	65
100,000 to 200,000	420 to 840	60
200,000 to 600,000	840 to 2500	55
600,000 to 1,200,000	2500 to 5000	50
1,200,000 to 1,800,000	5000 to 7500	45
Above 1,800,000	Above 7500	40

[a]This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, except that the second column has been added for convenience in interpreting the design load in terms of equivalent square feet of radiation.

[b]See also Time Analysis in Starting Heating Apparatus, by Ralph C. Taggert (A.S.H.V.E. TRANSACTIONS, Vol. 19, 1913); Report of A.S.H.V.E. Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930); Selecting the Right Size Heating Boiler, by Sabin Crocker (*Heating, Piping and Air Conditioning*, March, 1932).

[c]This table refers to hand fired solid fuel boilers. A factor of 25 per cent over design load is adequate when oil or gas are used as fuels.

[d]240 Btu per square foot.

Other things to be considered are:

5. Efficiency with hard or soft coal, gas or oil firing, as the case may be.

6. Grate area with hand-fired coal, or fuel burning rate with stokers, oil, or gas.

7. Combustion space in the furnace.

8. Type of heat liberation, whether continuous or intermittent, or a combination of both.

9. Miscellaneous items consisting of draft available, character of attendance, possibility of future extension, possibility of breakdown, headroom in the boiler room.

Radiation Load

The connected radiation (Item 1) is determined by calculating the heat losses in accordance with data given in Chapters 5, 6 and 7, and dividing by 240 to change to square feet of equivalent radiation as explained in Chapter 30. For hot water, the emission commonly used is 150 Btu per square foot, but the actual emission depends on the temperature of the medium in the heating units and of the surrounding air. (See Chapter 30).

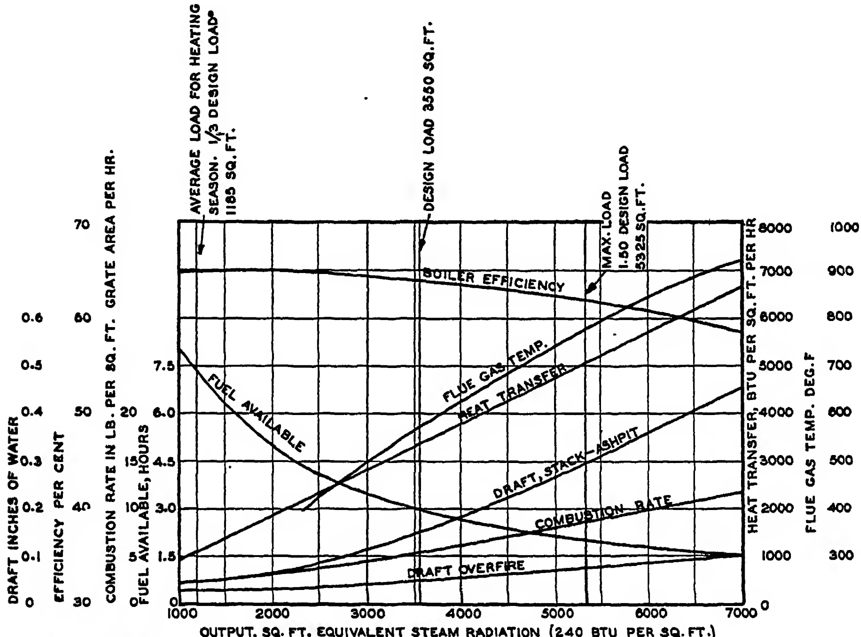
Although it is customary to use the actual connected load in equivalent square feet of radiation for selecting the size of boiler, this connected load

[8]See A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

usually represents a reserve in heating capacity to provide for infiltration in the various spaces of the building to be heated, which reserve, however, is not in use at all places at the same time, or in any one place at all times. For a further discussion of this subject see p. 101, Chapter 6.

Hot Water Supply Load

When the hot water supply (Item 2) is heated by the building heating boiler, this load must be taken into consideration in sizing the boiler. The



BOILER DATA—	GRATE AREA, SQ. FT.	16.0	FUEL—BITUMINOUS 3/4" LUMP		
	HEATING SURFACE, SQ. FT.	254	ANALYSIS—	VOLATILE MATTER	34.68%
	WEIGHT, LB.	9160		FIXED CARBON	55.44
	FUEL CAPACITY, LB.	651		ASH	9.87
	FUEL AVAILABLE, LB.	414		SULPHUR	2.86
	FUEL DEPTH, IN.	10		MOISTURE	3.00
				BTU PER LB.	13,655

FIG. 1. TYPICAL PERFORMANCE CURVES FOR A 36-IN. CAST-IRON SECTIONAL STEAM HEATING BOILER, BASED ON THE A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

allowance to be made will depend on the amount of water heated and its temperature rise. A good approximation is to add 4 sq ft of equivalent radiation for each gallon of water heated per hour through a temperature range of 100 F. For more specific information, see Chapter 39.

Piping Tax (Item 3)

It is common practice to add a flat percentage allowance to the equivalent connected radiation to provide for the heat loss from bare and covered pipe in the supply and return lines. The use of a flat allowance of 25 per cent for steam systems and 35 per cent for hot water systems is

preferable to ignoring entirely the load due to heat loss from the supply and return lines, but better practice, especially when there is much bare pipe, is to compute the emission from both bare and covered pipe surface in accordance with data in Chapter 35. With direct radiation served by bare supply and return piping the percentages may be higher than those stated, while in the case of unit heaters where the output is concentrated in a few locations, the piping tax may be 10 per cent or less.

Warming-up Allowance

The warming-up allowance represents the load due to heating the boiler and contents to operating temperature and heating up cold radiation and piping. (See Item 4). The factors to be used for determining the allowance to be made should be selected from Table 2 and should be applied to the estimated design load as determined by Items 1, 2 and 3.

Performance Curves for Boiler Selection

In the selection of a boiler to meet the estimated load, the A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers recommends the use of performance curves based on actual tests conducted in accordance with the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code No. 3), similar to the typical curves shown in Fig. 1. It should be understood that performance data apply to test conditions and that a reasonable allowance should be made for decreased output resulting from soot deposit, poor fuel or inefficient attention.

Selection Based on Heating Surface and Grate Area

Where performance curves are not available, a good general rule for conventionally-designed boilers is to provide 1 sq ft of boiler heating surface for each 14 sq ft of equivalent radiation (240 Btu per square foot) represented by the design load consisting of connected radiation, piping tax and domestic water heating load. As stated in the section on Boiler Output, this is equivalent to allowing 10 sq ft of boiler heating surface per boiler horsepower. In this case it is assumed that the maximum load including the warming-up allowance will be provided for by operating the boiler in excess of the design load, that is, in excess of the 100 per cent rating on a boiler-horsepower basis.

Due to the wide variation encountered in manufacturers' ratings for boilers of approximately the same capacity, it is advisable to check the grate area required for heating boilers burning solid fuel by means of the following formula:

$$G = \frac{H}{C \times F \times E} \tag{1}$$

where

G = grate area, square feet.

H = required total heat output of the boiler, Btu per hour (see Selection of Boilers, p. 337).

C = combustion rate in pounds of dry coal per square foot of grate area per hour depending on the kind of fuel and size of boiler as given in Table 1.

F = calorific value of fuel, Btu per pound.

E = efficiency of boiler, usually taken as 0.60.

Example 1. Determine the grate area for a required heat output of the boiler of 500,000 Btu per hour, a combustion rate of 6 lb per hour, a calorific value of 13,000 Btu per pound, and an efficiency of 60 per cent.

$$G = \frac{500,000}{6 \times 13,000 \times 0.60} = 10.7 \text{ sq ft}$$

The boiler selected should have a grate area not less than that determined by Formula 1. With small boilers where it is desired to provide sufficient coal capacity for approximately an eight-hour firing period plus a 20 per cent reserve for igniting a new charge, more grate area may be required depending upon the depth of the fuel pot.

Selection of Gas-Fired Boilers

Gas-heating appliances should be selected in accordance with factors given in Table 1, Chapter 28, which include an allowance for heating-up cold radiation, and for the piping tax. These factors are for thermostatically-controlled systems; in case manual operation is desired, a warming-up allowance of 100 per cent is recommended by the *A.G.A.* A gas boiler selected by the use of the *A.G.A.* factors will be the minimum size boiler which can carry the load. From a fuel economy standpoint, it may be advisable to select a somewhat larger boiler and then throttle the gas and air adjustments as required. This will tend to give a low stack temperature with high efficiency and at the same time provide reserve capacity in case the load is underestimated or more is added in the future.

Conversions

In the case of a solid fuel boiler converted to gas burning, the heat units supplied in the gas should be approximately twice the connected heating load. A combustion efficiency of 75 per cent for a conversion installation would provide a boiler output of $2 \times 0.75 = 1.5$ times the connected load, which allows 50 per cent for piping tax and pick-up. The presumption for a conversion job is that the boiler already is installed and probably will not be made larger; therefore, it is a matter of setting a gas-burning rate to obtain best results with the available surface. The conversion of a coal or oil boiler to gas burning is accomplished much more rapidly than the reverse since but little furnace volume need be provided for the proper combustion of gas.

Other Considerations in Selection of Boilers

As it will usually be found that several boilers will meet the specifications, the final selection of the boiler may be influenced by other considerations, some of which are:

1. Dimensions of boiler.
2. Durability under service.
3. Convenience in firing and cleaning.
4. Adaptability to changes in fuel and kind of attention.
5. Height of water line.

In large installations, the use of several smaller boiler units instead of one larger one will obtain greater flexibility and economy by permitting the operation, at the best efficiency, of the required number of units according to the heat requirements.

Boiler rooms should, if possible, be situated at a central point with

respect to the building and should be designed for a maximum of natural light. The space in front of the boilers should be sufficient for firing, stoking, ash removal and cleaning or renewal of flues, and should be at least 3 ft greater than the length of the boiler firebox.

A space of at least 3 ft should be allowed on at least one side of every boiler for convenience of erection and for accessibility to the various dampers, cleanouts and trimmings. The space at the rear of the boiler should be ample for the chimney connection and for cleanouts and with large boilers the rear clearance should be at least 3 ft in width.

The boiler room height should be sufficient for the location of boiler accessories and for proper installation of piping. In general the ceiling height for small steam boilers should be at least 3 ft above the normal boiler water line. With vapor heating especially, the height above the boiler water line is of vital importance.

When steel boilers are used, space should be provided for the removal and replacement of tubes.

BOILER CONNECTIONS AND FITTINGS

The velocity of flow through the outlets of low pressure steam heating boilers should not exceed 15 to 25 fps if fluctuation of the water line and undue entrainment of moisture are to be avoided. Steam or water outlet connections preferably should be the full size of the manufacturers' tapping and should extend vertically to the maximum height available above the boiler. For gravity circulating steam heating systems, it is recommended that a Hartford Loop, described in Chapter 32, be utilized in making the return connection.

Particular attention should be given to *fitting connections* to secure conformity with the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers. Attention is called in particular to pressure gage piping, water gage connections and safety valve capacity.

Steam gages should be fitted with a water seal and a shut-off consisting of a cock with either a tee or lever handle which is parallel to the pipe when the cock is open. *Steam gage connections* should be of copper or brass when smaller than 1 in. *I.P.S.*[9] if the gage is more than 5 ft from the boiler connection, and also in any case where the connection is less than ½ in. *I.P.S.*

Each steam or vapor boiler should have at least one *water gage* glass and two or more *gage cocks* located within the range of the visible length of the glass. The water gage fittings or gage cocks may be direct connected to the boiler, if so located by the manufacturer, or may be mounted on a separate water column. No connections, except for combustion regulators, drains or steam gages, should be placed on the pipes connecting the water column and the boiler. If the water column or gage glass is connected to the boiler by pipe and fittings a cross, tee or equivalent, in which a cleanout plug or a drain valve and piping may be attached, should be placed in the water connection at every right-angle turn to facilitate cleaning. The water line in steam boilers should be carried at the level specified by the boiler manufacturer.

[9]*A.S.M.E.* Code, Identification of Piping Systems.

Safety valves should be capable of discharging all the steam that can be generated by the boiler without allowing the pressure to rise more than 5 lb above the maximum allowable working pressure of the boiler. This should be borne in mind particularly in the case of boilers equipped with mechanical stokers or oil burners where the amount of grate area has little significance as to the steam generating capacity of the boiler.

Where a *return header* is used on a cast-iron sectional boiler to distribute the returns to both rear tappings, it is advisable to provide full size plugged tees instead of elbows where the branch connections enter the return tappings. This facilitates cleaning sludge from the bottom of the boiler sections through the large plugged openings. An equivalent clean-out plug should be provided in the case of a single return connection.

Blow-off or drain connections should be made near the boiler and so arranged that the entire system may be drained of water by opening the drain cock. In the case of two or more boilers separate blow-off connections must be provided for each boiler on the boiler side of the stop valve on the main return connection.

Water service connections must be provided for both steam and water boilers, for refilling and for the addition of make-up water to boilers. This connection is usually of galvanized steel pipe, and is made to the return main near the boiler or boilers.

For further data on pipe connections for steam and hot-water heating systems, see Chapters 32 and 33 and the *A.S.M.E.* Boiler Construction Code for Low Pressure Heating Boilers.

Smoke Breeching and Chimney Connections. The breeching or smoke pipe from the boiler outlet to the chimney should be air-tight and as short and direct as possible, preference being given to long radius and 45-deg instead of 90-deg bends. The breeching entering a brick chimney should not project beyond the flue lining and where practicable it should be grouted up from the inside of the chimney. A thimble or sleeve grout usually is provided where the breeching enters a brick chimney.

Where a battery of boilers is connected into a breeching each boiler should be provided with a tight damper. The breeching for a battery of boilers should not be reduced in size as it goes to the more remote boilers. Good connections made to a good chimney will usually result in a rapid response by the boilers to demands for heat.

BOILER ERECTION, OPERATION, AND MAINTENANCE

The directions of the boiler manufacturer always should be read before the assembly or installation of any boiler is started, even though the contractor may be familiar with the boiler. All joints requiring boiler putty or cement which cannot be reached after assembly is complete must be finished as the assembly progresses.

The following physical precautions should be taken in all installations to prevent damage to the boiler:

1. There should be provided proper and convenient drainage connections for use if the boiler is not in operation during freezing weather.

2. Strains on the boiler due to movement of piping during expansion should be prevented by suitable anchoring of piping and by proper provision for pipe expansion and contraction.

3. Direct impingement of too intense local heat upon any part of the boiler surface, as with oil burners, should be avoided by protecting the surface with firebrick or other refractory material.

4. Condensation must flow back to the boiler as rapidly and uniformly as possible. Return connections should prevent the water from backing out of the boiler.

5. Automatic boiler feeders and low water cut-off devices which shut off the source of heat if the water in the boiler falls below a safe level are recommended for boilers mechanically fired.

Boiler Troubles

A complaint regarding boiler operation generally will be found to be due to one of the following:

1. *The boiler fails to deliver enough heat.* The cause of this condition may be: (*a*) poor draft; (*b*) poor fuel; (*c*) inferior attention or firing; (*d*) boiler too small; (*e*) improper piping; (*f*) improper arrangement of sections; (*g*) heating surfaces covered with soot; and (*h*) insufficient radiation installed.

2. *The water line is unsteady.* The cause of this condition may be: (*a*) grease and dirt in boiler; (*b*) water column connected to a very active section and, therefore, not showing actual water level in boiler; (*c*) boiler operating at excessive output.

3. *Water disappears from gage glass.* This may be caused by: (*a*) priming due to grease and dirt in boiler; (*b*) too great pressure difference between supply and return piping preventing return of condensation; (*c*) valve closed in return line; (*d*) connection of bottom of water column into a very active section or thin waterway; (*e*) improper connections between boilers in battery permitting boiler with excess pressure to push returning condensation into boiler with lower pressure.

4. *Water is carried over into steam main.* This may be caused by: (*a*) grease and dirt in boiler; (*b*) insufficient steam dome or too small steam liberating area; (*c*) outlet connections of too small area; (*d*) excessive rate of output; (*e*) water level carried higher than specified.

5. *Boiler is slow in response to operation of dampers.* This may be due to: (*a*) poor draft due to air leaks into chimney or breeching; (*b*) inferior fuel; (*c*) inferior attention; (*d*) accumulation of clinker on grate; (*e*) boiler too small for the load.

6. *Boiler requires too frequent cleaning of flues.* This may be due to: (*a*) poor draft; (*b*) smoky combustion; (*c*) too low a rate of combustion; (*d*) too much excess air in firebox causing chilling of gases.

7. *Boiler smokes through fire door.* This may be due to: (*a*) defective draft in chimney or incorrect setting of dampers; (*b*) air leaks into boiler or breeching; (*c*) gas outlet from firebox plugged with fuel; (*d*) dirty or clogged flues; (*e*) improper reduction in breeching size.

Cleaning Steam Boilers

All boilers are provided with flue clean-out openings through which the heating surface can be reached by means of brushes or scrapers. Flues of solid fuel boilers should be cleaned often to keep the surfaces free of soot or ash. Gas boiler flues and burners should be cleaned at least once a year. Oil burning boiler flues should be examined periodically to determine when cleaning is necessary.

The grease used to lubricate the cutting tools during erection of new piping systems serves as a carrier for sand and dirt, with the result that a scum of fine particles and grease accumulates on the surface of the water in all new boilers, while heavier particles may settle to the bottom of the boiler and form sludge. These impurities have a tendency to cause foaming, preventing the generation of steam and causing an unsteady water line.

This unavoidable accumulation of oil and grease should be removed by blowing off the boiler as follows: If not already provided, install a

surface blow connection of at least 1¼ in. nominal pipe size with outlet extended to within 18 in. of the floor or to sewer, inserting a valve in line close to boiler. Bring the water line to center of outlet, raise steam pressure and while fire is burning briskly open valve in blow-off line. When pressure recedes close valve and repeat process adding water at intervals to maintain proper level. As a final operation bring the pressure in the boiler to about 10 lb close blow-off, draw the fire or stop burner and open drain valve. After boiler has cooled partly fill and flush out several times before filling it to proper water level for normal service. The use of acids, alkalis and salts for cleaning is not favored by boiler manufacturers because of difficulty of complete removal and the possibility of subsequent injury.

Insoluable compounds have been developed which are effective, but special instructions on the proper cleaning compound and directions for its use in a boiler, as given by the boiler manufacturer, should be carefully followed.

When soda ash solution is to be used the procedure is to add about 5 lb of soda ash for each 1000 sq ft of connected radiation. Fill the boiler with water until it just overflows from the surface blow outlet pipe and then fire sufficiently to raise the water temperature to the boiling point without getting up steam pressure. Crack the boiler feed valve so that a steady trickle will run out of the overflow pipe. Allow the boiler to simmer from 2 to 4 hours. At the end of this time the grease and sediment should have passed off through the overflow pipe or loosened sufficiently to drain off through the bottom blow. Extinguish the fire—preferably by letting it burn out and then dumping any live coals into the ashpit where water can be applied with a hose—and open the bottom blow wide. Rinse with fresh water and refill to the normal water level. If the water in the gage glass then does not show clear, repeat the process using a stronger soda ash solution and boiling for a longer time. It sometimes is necessary to repeat this process several times to completely rid the boiler of grease. Failure to thoroughly eliminate grease usually results in an unsteady water line and danger of damaging the boiler through having the crown-sheet uncovered.

It is common practice when starting new installations to discharge heating returns to the sewer during the first week of operation. This prevents the passage of grease, dirt or other foreign matter into the boiler and consequently may avoid the necessity of cleaning the boiler. During the time the returns are being passed to the sewer, the feed valve should be cracked sufficiently to maintain the proper water level in the boiler.

Care of Idle Heating Boilers

Heating boilers are often seriously damaged during summer months due chiefly to corrosion resulting from the combination of sulphur from the fuel with the moisture in the cellar air. At the end of the heating season the following precautions should be taken:

1. All heating surfaces should be cleaned thoroughly of soot, ash and residue, and the heating surfaces of steel boilers should be given a coating of lubricating oil on the fire side.

2. All machined surfaces should be coated with oil or grease.

3. Connections to the chimney should be cleaned and in case of small boilers the pipe should be placed in a dry place after cleaning.

4. If there is much moisture in the boiler room, it is desirable to drain the boiler to prevent atmospheric condensation on the heating surfaces of the boiler when they are below the dew-point temperature. Due to the hazard of some one inadvertently building a fire in a dry boiler, however, it is safer to keep the boiler filled with water. A hot water system usually is left filled to the expansion tank.

5. The grates and ashpit should be cleaned.

6. Clean and repack the gage glass if necessary.

7. Remove any rust or other deposit from exposed surfaces by scraping with a wire brush or sandpaper. After boiler is thoroughly cleaned, apply a coat of preservative paint where required to external parts normally painted.

8. Inspect all accessories of the boiler carefully to see that they are in good working order. In this connection, oil all door hinges, damper bearings and regulator parts.

BOILER INSULATION

Insulation for cast-iron boilers is of two general types: (1) plastic material or blocks wired on, cemented and covered with canvas or duck; and (2) blocks, sheets or plastic material covered with a metal jacket furnished by the boiler manufacturer. Self-contained steel firebox boilers usually are insulated with blocks, cement and canvas, or rock wool blankets; HRT boilers are brick set and do not require insulation beyond that provided in the setting. It is essential that the insulation on a boiler and adjacent piping be of non-combustible material as even slow-burning insulation constitutes a dangerous fire hazard in case of low water in the boiler.

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Chapter 26

CHIMNEYS

Natural Draft, Mechanical Draft, Characteristics of Natural Draft Chimneys, Determining Chimney Sizes, General Equation, Chimney Construction, Chimneys for Gas Heating

DRAFT, in general, may be defined as the pressure difference between the atmospheric pressure and that at any part of an installation through which the gases flow. Since a pressure difference implies a head, draft then is a static force. While no element of motion is inferred, yet motion in the form of circulation of gases throughout an entire boiler plant installation is the direct result of draft. This motion is due to the pressure difference, or unbalanced pressure, which compels the gases to flow.

Draft is often classified into two kinds according to whether it is created thermally or artificially, *viz*, (1) natural or thermal draft, and (2) artificial or mechanical draft.

Natural Draft

Natural draft is the difference in pressure produced by the difference in weight between the relatively hot gases inside a natural draft chimney and an equivalent column of the cooler outside air, or atmosphere. Natural draft, in other words, is an unbalanced pressure produced thermally by a natural draft chimney as the pressure transformer and a temperature difference. The intensity of natural draft depends, for the most part, upon the height of the chimney above the grate bar level and also the temperature difference between the chimney gases and the atmosphere.

A typical natural draft system consists essentially of a relatively tall chimney built of steel, brick or reinforced concrete, operating with the relatively hot gases which have passed through the boilers and accessories and from which all of the heat has not been extracted. Hot gases are an essential element in the operation of a natural draft system.

A natural-draft chimney performs the two-fold service of assisting in the creation of draft by aspiration and also of discharging the gases at an elevation sufficient to prevent them from becoming a nuisance.

Natural draft is quite advantageous in installations where the total loss of draft due to resistances is relatively low and also in plants which have practically a constant load and whose boilers are seldom operated above their normal rating. Natural draft systems have been, and are still being, employed in the operation of large plants during the periods when the boilers are operated only up to their normal rating. When the rate of operation is increased above their normal rating, some form of mechanical

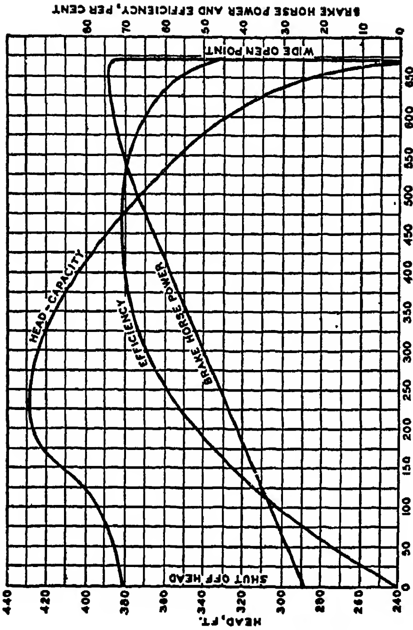


FIG. 2. OPERATING CHARACTERISTICS OF TYPICAL CENTRIFUGAL PUMP

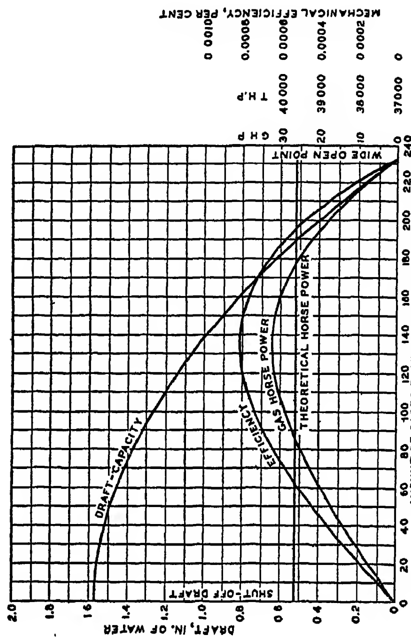


FIG. 3. OPERATING CHARACTERISTICS OF TYPICAL NATURAL DRAFT CHIMNEY

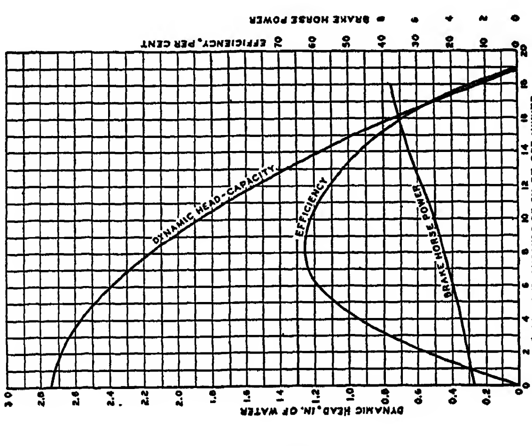


FIG. 1. GENERAL OPERATING CHARACTERISTICS OF TYPICAL INDUCED DRAFT FAN

draft is employed as an auxiliary to overcome the increased resistances or draft losses. Natural draft systems are used almost exclusively in the smaller size plants where the amount of gases generated is relatively small and it would be expensive to install and operate a mechanical draft system.

The principal advantages of natural draft systems may be summarized as follows: (1) simplicity, (2) reliability, (3) freedom from mechanical parts, (4) low cost of maintenance, (5) relatively long life, (6) relatively low depreciation, and (7) no power required to operate. The principal disadvantages are: (1) lack of flexibility, (2) irregularity, (3) affected by surroundings, and (4) affected by temperature changes.

Mechanical Draft

Artificial draft, or mechanical draft, as it is more commonly called, is a difference in pressure produced either directly or indirectly by a forced draft fan, an induced draft fan or a Venturi chimney as the pressure transformer. The intensity of mechanical draft is dependent for the most part upon the size of the fan and the speed at which it is operated. The element of temperature does not enter into the creation of mechanical draft and therefore its intensity, unlike natural draft, is independent of the temperature of the gases and the atmosphere. Mechanical draft includes the induced and Venturi types of draft systems in which the pressure difference is the result of a suction and also the forced draft system in which the pressure difference is the result of a blowing. Mechanical draft systems tend to produce a vacuum or a plenum, according as the system used in its production creates a pressure difference below, or above, atmospheric pressure, respectively. A mechanical draft system may be used either in conjunction with, or as an adjunct to, a natural draft system.

CHARACTERISTICS OF NATURAL DRAFT CHIMNEYS

In order to analyze the performance of a natural draft chimney, it is advantageous to compare its general operating characteristics with those of a centrifugal pump and also a centrifugally-induced draft fan, there being a close similarity among the three. Figs. 1, 2 and 3 show the general operating characteristics of a typical centrifugally-induced draft fan, a typical centrifugal pump, and a typical natural draft chimney, respectively. The draft-capacity curve of the chimney corresponds to the head-capacity curve of the pump and also to the dynamic-head capacity of the fan; the efficiency curve of the chimney to the efficiency curves of the pump and fan; and the gas horsepower curve of the chimney to the brake horsepower curves of the pump and fan.

When the gases in the chimney are stationary, the draft created is termed the *theoretical draft.* When the gases are flowing, the theoretical intensity is diminished by the draft loss due to friction, the difference between the two being termed the *available draft.* The general equation for the available draft intensity of a natural draft chimney with a circular section is as follows:

$$D_a = 2.96HB_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.00126W^2T_cfL}{D^5B_oW_c} \qquad (1)$$

where

D_a = available draft, inches of water.

H = height of chimney above grate bars, feet.

B_0 = barometric pressure corresponding to altitude, inches of mercury.

W_0 = unit weight of a cubic foot of air at 0 deg Fahrenheit and sea level atmospheric pressure, pounds per cubic foot.

W_c = unit weight of a cubic foot of chimney gases at 0 deg Fahrenheit and sea level atmospheric pressure, pounds per cubic foot.

T_0 = absolute temperature of atmosphere, degrees Fahrenheit.

T_c = absolute temperature of chimney gases, degrees Fahrenheit.

W = amount of gases generated in the combustion chamber of the boiler and passing through the chimney, pounds per second.

f = coefficient of friction.

L = length of friction duct of the chimney, feet.

D = minimum diameter of chimney, feet.

The first term of the right hand expression of Equation 1 represents the theoretical draft intensity and the second term, the loss due to friction.

Example 1. Determine the available draft of a natural draft chimney 200 ft in height and 10 ft in diameter operating under the following conditions: atmospheric temperature, 62 F; chimney gas temperature, 500 F; sea level atmospheric pressure, B_0 = 29.92 in. of mercury; atmospheric and chimney gas density, 0.0863 and 0.09, respectively; coefficient of friction, 0.016; length of friction duct, 200 ft; and discharging 100 lb of gases per second.

Substituting these values in Equation 1 and reducing:

$$D_a = 2.96 \times 200 \times 29.92 \times \left(\frac{0.0863}{522} - \frac{0.09}{960}\right) - \frac{0.00126 \times 100^2 \times 960 \times 0.016 \times 200}{10^5 \times 29.92 \times 0.09}$$

$$= 1.27 - 0.14 = 1.13 \text{ in.}$$

Fig. 4 shows the variation in the available draft of a typical 200 ft by 10 ft chimney operating under the general conditions noted in Example 1. When the chimney is under static conditions and no gases are flowing, the available draft is equal to 1.27 in. of water, the theoretical intensity. As the amount of gases flowing increases, the available intensity decreases until it becomes zero at a gas flow of 297 lb per second at which point the draft loss due to friction is equal to the theoretical intensity. The draft-capacity curve corresponds to the head-capacity curve of centrifugal pump characteristics and the dynamic-head-capacity curve of a fan. The point of maximum draft and zero capacity is called shut-off draft, or point of impending delivery, and corresponds to the point of shut-off head of a centrifugal pump. The point of zero draft and maximum capacity is called the wide open point and corresponds to the wide open point of a centrifugal pump. A set of operating characteristics may be developed for any size chimney operating under any set of conditions by substituting the proper values in Equation 1 and then plotting the results in the manner shown in Fig. 4.

The efficiency of a natural draft chimney is the thermodynamical ratio of the energy output to the energy input. The energy output is the total

work done by the chimney in moving the gases and corresponds to the water horsepower of a centrifugal pump, or the total work done by a fan in moving the gases. The energy input is equal to the theoretical amount of power generated by the chimney and corresponds to the power input of the driving unit of a centrifugal pump or a fan. The thermodynamical efficiency is given by the equation:

$$E_t = \frac{K_a W D_a}{A\sqrt{H}} \tag{2}$$

where

K_a = a constant depending upon the temperature of the gases, the atmospheric temperature, the elevation of the plant, and the density and specific heat of the gases. For average operating conditions, K_a = 0.0065.

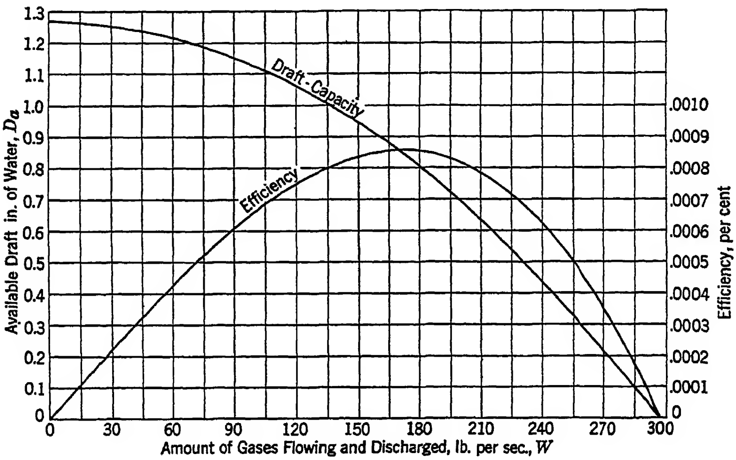


FIG. 4. TYPICAL SET OF OPERATING CHARACTERISTICS OF A NATURAL DRAFT CHIMNEY

Fig. 4 shows the variation in the efficiency of the chimney under consideration for the operating conditions noted. This curve rises from zero at shut-off draft to a maximum for a certain draft and its corresponding capacity and then drops again to zero at the wide open point. The point of maximum efficiency is located by the point on the draft-capacity curve equal to two-thirds of the theoretical draft intensity. In Example 1 the maximum efficiency is at an available draft intensity of ⅔ × 1.27 = 0.85 in. of water and the corresponding capacity of 175 lb per second.

The efficiency curve of a natural draft chimney corresponds to the efficiency curves of a centrifugal pump and a fan and serves the same general use in that it locates the region of most economical operation. In substituting the values for the various factors in Equation 1, care should be exercised that the selections be as near the actual conditions as is practically possible. The following notes will serve as a guide for these selections:

1. The *barometric pressure* varies inversely as the altitude of the plant above sea level. Fig. 5 gives the barometric pressure corresponding to various elevations as computed from the equation:

$$E_1 = 62{,}737 \log \frac{29.92}{B_o} \tag{3}$$

where

E_1 = altitude of plant above sea level, feet.

In general, the barometric pressure decreases approximately 0.1 in. of mercury per 100 ft increase in elevation.

2. The *unit weight of a cubic foot* of chimney gases at 0 deg Fahrenheit and sea level barometric pressure is given by the equation:

$$W_C = 0.131 CO_2 + 0.095\, O_2 + 0.083\, N_2 \tag{4}$$

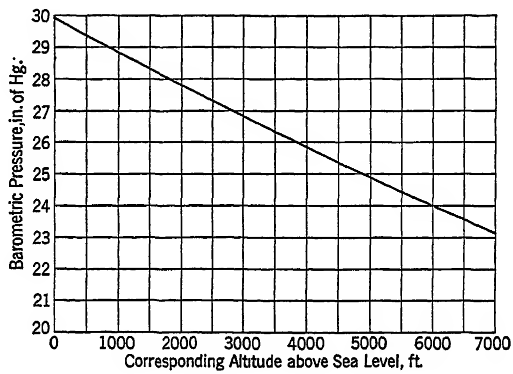


FIG. 5. RELATION BETWEEN BAROMETRIC PRESSURE AND ALTITUDE

in which CO_2, O_2 and N_2 represent the percentages of the parts by weight of the carbon dioxide, oxygen and nitrogen content, respectively, of the gas analysis. For ordinary operating conditions, the value of W_C may be assumed at 0.09.

3. The *atmospheric temperature* is the actual observed temperature of the outside air at the time the analysis of the operating chimney is made. The mean atmospheric temperature in the temperate zone is approximately 62 F.

4. The *chimney gas temperature* does not vary appreciably from the gas temperature as it leaves the breeching and enters the chimney. For average operating conditions, the chimney gas temperature will vary between 500 F and 650 F except in the case when economizers and recuperators are used, when the temperature will vary between 300 F and 450 F. If a chimney has been properly constructed, properly lined and has no air infiltration due to open joints, the temperature of the gases throughout the chimney will not differ appreciably from the foregoing figures. In most up-to-date heating plants, the temperature may be read from instruments or ascertained from a pyrometer.

5. The *coefficient of friction* between the chimney gases and a sooted surface has been found to be approximately 0.016. This factor, of course, will be much less for a new unlined steel stack than for a brick or brick-lined chimney, but in time the inside surface of all chimneys regardless of the material of which they are constructed becomes covered with a layer of soot and the coefficient of friction should be the same for all types of chimneys.

6. The *length of the friction duct* is the vertical distance between the bottom of the breeching opening and the top of the chimney. Ordinarily this distance is approximately equal to the height of the chimney above the grate level.

7. The *amount of gases flowing and being discharged* is, of course, equal to the amount of gases generated in the combustion chamber of the boiler. The total products of combustion may be computed from the equation:

$$W = \frac{C_g G W_{tp}}{3600} \tag{5}$$

where

C_g = pounds of fuel burned per square foot of grate surface per hour.
G = total grate surface of boilers, square feet.
W_{tp} = total weight of products of combustion per pound of fuel.

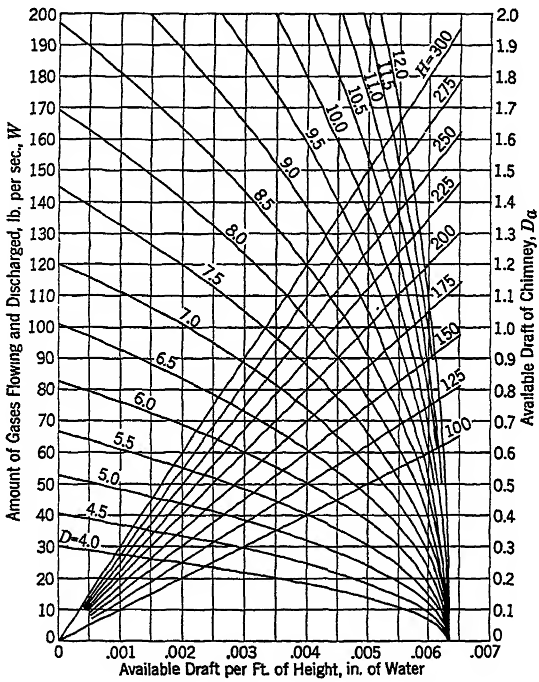


FIG. 6. CHIMNEY PERFORMANCE CHART

Fig. 6 is a typical chimney performance chart giving the available draft intensities for various amounts of gases flowing and sizes of chimney. This chart is based on an atmospheric temperature of 62 F, a chimney gas temperature of 500 F, a unit chimney gas weight of 0.09 lb per cubic foot, sea level atmospheric pressure, a coefficient of friction of 0.016, and a friction duct length equal to the height of the chimney above the grate level. These curves may be used for general operating conditions. For specific operating conditions, a new chart should be constructed from Equation 1.

It has been the usual custom, and still is to a lamentably great extent, to select the required size of a natural draft chimney from a table of

chimney sizes based only on boiler horsepowers. After the ultimate horsepower of the projected plant had been determined, the chimney size in the table corresponding to this figure was then selected as the proper size required. Generally, no further attempt was made to determine if the height thus selected was sufficient to help create the required draft demanded by the entire installation, or the diameter sufficiently large to enable the chimney quickly, efficiently and economically to dispose of the gases. Since the operating characteristics of a natural draft chimney are similar in all respects to those of a centrifugal pump, or a centrifugal fan, it is no more possible to select a proper size chimney from such a table, even with correction factors appended, than it is to select the proper size pump from tables based only on the amount of water to be delivered.

DETERMINING CHIMNEY SIZES

The required diameter and height of a natural draft chimney are given by the following equations:

$$H = \frac{D_r}{2.96B_o\left(\frac{W_o}{T_o} - \frac{W_c}{T_c}\right) - \frac{0.184fW_cB_oV^2}{T_cD}} \tag{6}$$

$$D = 0.288\sqrt{\frac{WT_c}{B_oW_cV}} \tag{7}$$

where

H = required height of chimney above grate bar level, feet.
D = required minimum diameter of chimney, feet.
V = chimney gas velocity, feet per second.
D_r = total required draft demanded by the entire installation outside of the chimney, inches of water.

Equations 6 and 7 give the required size of a natural draft chimney with all of the operating factors taken into consideration. Values for all of the factors with the exception of the chimney gas velocity may be either observed or computed. It is, of course, necessary to assume an arbitrary value for the velocity in order to arrive at some definite size. For any one set of operating conditions there will be as many sizes of chimney as there are values of reasonable velocities to assume. Of the number of sizes corresponding to the various assumed velocities, there is one size which will cost least. Since the cost of a chimney structure, regardless of the kind of material used in the construction, varies as the volume of material in the structure, the cost criterion then may be represented by the approximate equation:

$$Q = \pi tHD \tag{8}$$

where

Q = volume of material, cubic feet.
t = average wall thickness, feet.

For all practical purposes, the value of πt may be taken as a constant regardless of the size of the structure. Hence, in general, the volume, and consequently the cost, of a chimney structure may be based on the factor

HD as a criterion. Therefore, the value of the chimney gas velocity which will result in the least value of *HD* for any one set of operating conditions will produce a structure whose cost will be least and, as a result, will be the most economical to use.

The problem at hand is to deduce an equation for the chimney gas velocity which will result in a combination of a height and a diameter whose product *HD* will be least. The solution is obtained by equating the

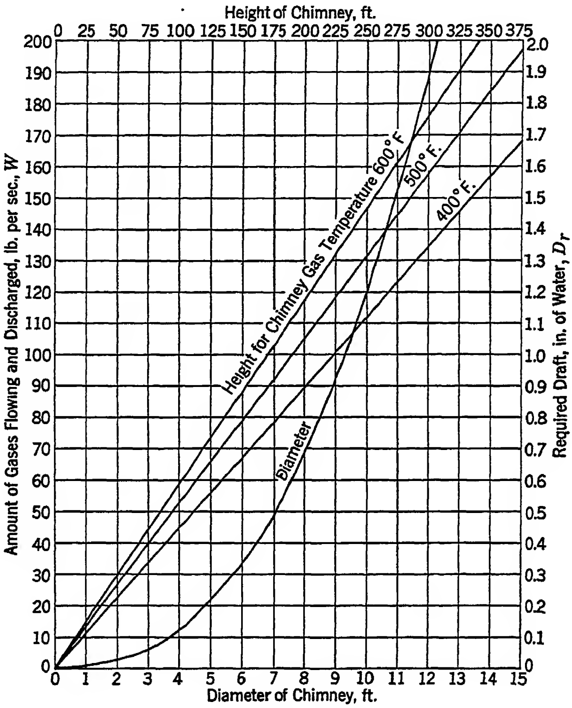


FIG. 7. ECONOMICAL CHIMNEY SIZES

product of Equations 3 and 4 to *HD*, differentiating this product with respect to *V* and equating the resulting expression to zero. This procedure results in the following expression:

$$V_e = \left(\frac{0.772 T_c \left(\frac{W_o}{T_o} - \frac{W_c}{T_c} \right) \sqrt{\frac{W T_c}{B_o W_c}}}{f W_c} \right)^{2/5} \qquad (9)$$

where V_e = economical chimney gas velocity, feet per second.

Equation 9 gives the economical velocity of the chimney gases for any set of operating conditions, and represents the velocity which will result in a chimney the size of which will cost less than that of any other size as determined by any other velocity for the same operating conditions. After the value of the economical velocity has been determined, the

corresponding height and diameter can then be determined from Equations 6 and 7, respectively, and the economical size will then be attained. Equations 6, 7 and 9 may be simplified considerably for average operating conditions in an average size steam plant by assuming the following conditions:

Average chimney gas temperature, 500 F .... $T_c = 960$
Mean atmospheric temperature, 62 F .... $T_o = 522$
Average coefficient of friction, 0.016 .... $f = 0.016$
Average chimney gas density, 0.09 .... $W_c = 0.09$
Sea level elevation with barometer of 29.92 .... $B_o = 29.92$

Substituting these values in Equations 9, 7 and 6, respectively, and reducing:

$$V_e = 13.7W^{1/5} \tag{10}$$

$$D = 1.5W^{2/5} \tag{11}$$

$$H = 190D_r \tag{12}$$

Fig. 7 gives the economical chimney sizes for various amounts of gases flowing and for required draft intensities as computed from Equations 10, 11 and 12. They are based on the operating factors used in reducing Equations 6, 7 and 9 to their simpler form. The sizes shown by the curves in the chart should be used for general operating conditions only, or for installations where the required data necessary for an exact determination are difficult or impossible to secure. Whenever it is possible to secure accurate data, or the anticipated operating conditions are fairly well known, the required size should be determined from Equations 6, 7 and 9. The recommended minimum inside dimensions and heights of chimneys for small and medium size installations are given in Table 1.

GENERAL EQUATION

The general draft equation for a steam producing plant may be stated as follows:

$$D_t - h_f = h_F + h_B + h_{Bd} + h_C + h_{Br} + h_V + h_O + h_E + h_R \tag{13}$$

where

D_t = theoretical draft intensity created by pressure transformer, inches of water.
h_f = draft loss due to friction in pressure transformer, inches of water.
h_F = draft loss through the fuel bed, inches of water.
h_B = draft loss through the boiler and setting, inches of water.
h_{Br} = draft loss through the breeching, inches of water.
h_V = draft loss due to velocity, inches of water.
h_{Bd} = draft loss due to bends, inches of water.
h_C = draft loss due to contraction of opening, inches of water.
h_O = draft loss due to enlargement of opening, inches of water.
h_E = draft loss through the economizer, inches of water.
h_R = draft loss through recuperators, regenerators or air heaters, inches of water.

The left hand member of Equation 13 represents the total amount of available draft created by the pressure transformer, that is, the natural

TABLE 1. RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES†

WARM AIR FURNACE CAPACITY IN SQ IN. OF LEADER PIPE	STEAM BOILER CAPACITY SQ FT OF RADIATION	HOT WATER HEATER CAPACITY SQ FT OF RADIATION	NOMINAL DIMENSIONS OF FIRE CLAY LINING IN INCHES	RECTANGULAR FLUE		ROUND FLUE		HEIGHT IN FT ABOVE GRATE
				Actual Inside Dimensions of Fire Clay Lining in Inches	Actual Area Sq In.	Inside Diameter of Lining in Inches	Actual Area Sq In.	
790	590	973	8½x13	7 x11½	81			35
1000	690	1,140				10	79	
	900	1,490	13x13	11¼x11¼	127			
	900	1,490	8½x18	6¾x16¼	110			
	1,100	1,820				12	113	40
	1,700	2,800	13x18	11¼x16¼	183			
	1,940	3,200				15	177	
	2,130	3,520	18x18	15¾x15¾	248			
	2,480	4,090	20x20	17¼x17¼	298			45
	3,150	5,200				18	254	50
	4,300	7,100				20	314	
	4,600	7,590	20x24	17x21	357			
	5,000	8,250	24x24	21x21	441			55
	5,570	9,190		24x24*	576			60
	5,580	9,200				22	380	
	6,980	11,500				24	452	65
	7,270	12,000		24x28*	672			
	8,700	14,400		28x28*	784			
	9,380	15,500				27	573	
	10,150	16,750		30x30*	900			
	10,470	17,250		28x32*	896			

*Dimensions are for unlined rectangular flues.

†This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (Edition of 1929).

draft chimney, Venturi chimney, or fan, and is equal to the theoretical intensity less the internal losses incidental to operation. The right hand member represents the sum of all of the various losses of draft throughout the entire boiler plant installation outside of the pressure transformer itself. The left hand member expresses the available intensity and is analogous to the head developed by a centrifugal pump in a water works system, while the right hand member expresses the required draft intensity and is analogous to the total dynamic head in a water works system. For a general circulation of gases then

$$D_a = D_r \tag{14}$$

where

D_a = available draft intensity, inches of water.
D_r = required draft, inches of water

The *draft loss through the fuel bed* (h_F), or the amount of draft required to effect a given or required rate of combustion, varies between wide limits and represents the greater portion of the required draft. In coal-fired installations, the draft loss through the fuel bed is dependent upon the following factors: (1) character and condition of the fuel, clean or dirty; (2) percentage of ash in the fuel; (3) volume of interstices in the fuel bed,

coarseness of fuel; (4) thickness of the fuel bed, rate of combustion; (5) type of grate or stoker used; (6) efficiency of combustion.

There is a certain intensity of draft with which the best results will be obtained for every kind of coal and rate of combustion. Fig. 8 gives the intensity of draft, or the vacuum in the combustion chamber required to burn various kinds of coal at various rates of combustion. Expressed in other words, these curves represent the amount of draft required to force the necessary amount of air through the fuel bed in order to effect various rates of combustion. It will be noted that the amount of draft increases as the percentage of volatile matter diminishes, being comparatively low for the lower grades of bituminous coals and highest for the high grades and small sizes of anthracites. Also, when the interstices of the coal are large and the particles are not well broken up, as with bituminous coals,

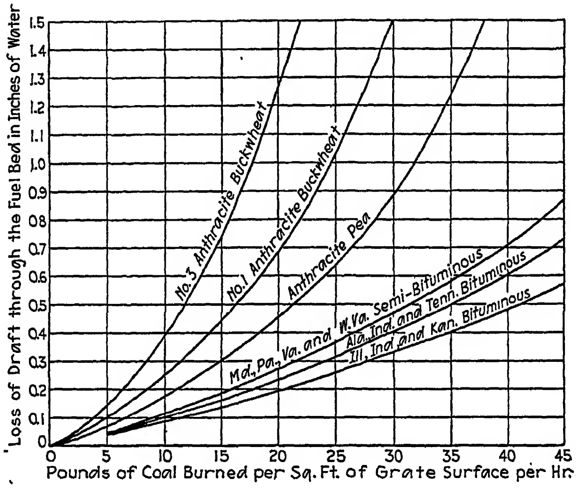


FIG. 8. DRAFT REQUIRED AT DIFFERENT RATES OF COMBUSTION FOR VARIOUS KINDS OF COAL

much less draft is required than when the particles are small and are well broken up, as with bituminous slack and the small sizes of anthracites. In general, the draft loss through the fuel bed increases as: (1) the percentage of volatile matter diminishes; (2) the percentage of fixed carbon increases; (3) the thickness of the bed increases; (4) the percentage of ash increases; (5) the volume of the interstices diminishes.

In making the preliminary assumptions for the draft loss through the fuel bed, due allowances should be made for a possible future change in the grade of fuel to be burned and also in the rate of combustion. A value should be selected for this loss which will represent not only the highest rate of combustion which will be encountered, but also the grade of coal which has the greatest resistance through the fuel bed and which may be burned at a later date.

In powdered-fuel and oil-fired installations, there will be no draft loss

through the fuel bed since there is none and, consequently, this factor becomes zero in the general draft equation. All other factors being constant, the height of the chimney in installations of this character will be less than the height in coal-fired installations, and in the case of mechanical draft installations the driving units need not be as large since the head against which the fan is to operate is not as great in the former as in the latter.

The *draft loss through the boiler and setting* (h_B) also varies between wide limits and, in general, depends upon the following factors:

1. Type of boiler.
2. Size of boiler.
3. Rate of operation.
4. Arrangement of tubes.
5. Arrangement of baffles.
6. Type of grate.
7. Design of brickwork setting.
8. Excess air admitted.
9. Location of entrance into breeching.

Curves showing the draft loss through the boiler are usually based on the load or quantity of gases passing through the boiler, expressed in terms of percentage of normal rate of operation. Owing to the great variety of boilers of different designs and the various schemes of baffling, it is impossible to group together a set of curves for the draft loss through the boiler which may even be used generally. It is therefore necessary to secure this information from the manufacturer of the particular type of boiler and baffle arrangement under consideration.

When a boiler is installed and in operation, the draft loss depends upon the amount of gases flowing through it. This, in turn, depends upon the proportion of excess air admitted for combustion. The amount of excess air is measured by the CO_2 content; the less the amount of CO_2, the greater the amount of excess air and hence the greater the draft loss.

The loss of draft through the boiler will vary directly as the size of the boiler and the length of the gas passages within. The loss also varies as the number of tubes high, but not in a direct ratio inasmuch as the loss due to the reversal of flow at the ends of the baffles remains constant regardless of the height of the boiler. The arrangement of the tubes, whether the gases flow parallel to or at right angles to the tubes, has an appreciable effect on the loss. The arrangement of the baffles influences the draft loss greatly, the loss through a boiler with five passes being greater than the loss through one of three or four passes. A poor design and a rough condition of the brickwork will increase the loss greatly, whereas a proper design and a smooth condition will keep the loss at a minimum. The loss through the boiler will be less when the breeching entrance is located at or near the top of the boiler than when it is located at or near the bottom since the gases have a shorter distance to travel in the former instance.

The *draft loss through the breeching* (h_{Br}) is given by the general equation:

$$h_{Br} = \frac{0.000194 W^2 T_C f L}{A^2 B_O W_C C_{br}} \tag{15}$$

where

W = the amount of gases flowing, pounds per second.
T_c = absolute temperature of breeching gases, degrees Fahrenheit.
f = coefficient of friction.
L = length of breeching, feet.
A = area of breeching, square feet.
B_o = atmospheric pressure corresponding to altitude, inches of mercury.
W_c = weight of a cubic foot of breeching gases at 0 F and sea level atmospheric pressure, pounds per cubic foot.
C_{br} = hydraulic radius of breeching section.

It has been the general custom to *lump off* the intensity of the breeching loss at 0.10 in. of water per 100 ft of breeching length regardless of its size or shape or the amount and temperature of the gases flowing through it. This practice is hazardous and has no more foundation in fact than that of determining the friction head in a water works system without taking into consideration the size of the pipe or the amount of water flowing through it. When the length of the breeching is relatively short, any variation in any one of the factors in the equation will have no appreciable effect on the draft loss. However, when the breeching is relatively long, the draft loss is affected greatly by the various factors, particularly by the size and shape as well as by the weight of gases flowing.

The *draft loss due to velocity* (h_V) is given by the equation

$$h_V = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \tag{16}$$

and represents the amount of draft required to accelerate the gases from zero velocity to the velocity at which the gases are flowing, or in other words, from a static gas condition of zero flow to the amount of gases flowing throughout the installation. This loss corresponds to the velocity head in water works systems.

The *draft loss due to bends* (h_{Bb}) is equivalent to the loss due to the velocity head for a 90-deg bend. In changing direction of flow, the gas velocity decreases to zero with a loss of velocity head and then increases to its proper value at the expense of a loss in pressure head, the net result being a loss in pressure head equal to the velocity head at the bend. This loss is given by the equation:

$$h_{Bd} = \frac{0.000194 W^2 T_c}{A^2 B_o W_c} \tag{17}$$

The friction at a right-angle bend is sometimes expressed as the equivalent of a straight length of flue of a certain length for a certain diameter, similar to the procedure used in estimating the loss due to bends in piping systems conducting water. Most flues, however, particularly breechings, are built square or rectangular in section and no general equation based on the shape of the flue can be conveniently expressed.

The *draft loss due to sudden contraction of an area* (h_C) is given by the equation:

$$h_C = \frac{0.000194 K_C W^2 T_C}{A_s^2 B_O W_C} \tag{18}$$

where

K_C = coefficient of sudden contraction based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section.

A_s = area of the smaller section.

When the flue or passage through which the gases flow is suddenly contracted, a considerable portion of the static head in the larger section is converted into velocity head and a draft loss of some consequence, particularly in a short breeching, takes place. A sudden contraction should always be avoided where possible. At times, however, due to obstructions or limited head-room, it is necessary to alter the size of the breeching, but a sudden contraction may be avoided by gradually decreasing the area over a length of several feet.

The *draft loss due to a sudden enlargement of an area* (h_O) is given by the equation:

$$h_O = \frac{0.000194 K_O W^2 T_C}{A_s^2 B_O W_C} \tag{19}$$

where

K_O = coefficient of sudden enlargement based on $\frac{A_s}{A_1}$, the ratio of the areas of the smaller to the larger section.

When the flue or passage through which the gases flow is suddenly enlarged, a portion of the velocity head is converted into static head in the larger section and, like the loss due to sudden contraction, a loss of some consequence, particularly in short breechings, takes place. A sudden enlargement in a breeching may be avoided by gradually increasing the area over a length of several feet. In large masonry chimneys, the area of the flue at the region of the breeching entrance is considerably larger than the area of the breeching at the chimney, and a sudden enlargement exists.

The *draft loss through the economizer* (h_E) should be obtained from the manufacturer but for general purposes it may be computed from the following general equation:

$$h_E = \frac{6.6 W_n^2 N T_C}{10^{12}} \tag{20}$$

where

W_n = pounds of gases flowing per hour per linear foot of pipe in each economizer section.

N = number of economizer sections.

An economizer in a steam plant affects the draft in two ways, (1) it offers a resistance to the flow of gases, and (2) it lowers the average chimney gas temperature, thereby decreasing the available intensity. In the case of a natural draft installation, both of these factors result in a relative increase in the height of the chimney and, in the case of a large

plant, they may add as much as 20 or 30 ft to the height. The decrease in the temperature of the gases after they have passed through the economizer has an extremely important effect on the performance of a natural draft chimney and also upon the performance of a fan.

CHIMNEY CONSTRUCTION

For general data on the construction of chimneys reference should be made to the Standard Ordinance for Chimney Construction of the *National Board of Fire Underwriters.* Briefly summarized, these provisions are as follows for heating boilers and furnaces:

The construction, location, height and area of the chimney to which a heating boiler or warm-air furnace is connected affect the operation of the entire heating system. Most residence chimneys are built of brick and may be either lined or unlined, but in either case the *walls must be air-tight* and there should be *only one smoke opening* into the chimney. Cleanout, if provided, must be *absolutely air-tight* when closed.

The walls of brick chimneys shall be not less than 3¾ in. thick (width of a standard size brick) and shall be lined with fire-clay flue lining. Fire-clay flue linings shall be manufactured from suitable refractory clay, either natural or compounded, and shall be adapted to withstand high temperatures and the action of flue gases. They shall be of standard commercial thickness, but not less than ¾ in. All fire-clay flue linings shall meet the standard specification of the *Eastern Clay Products Association.* The flue sections shall be set in special mortar, and shall have the joints struck smooth on the inside. The masonry shall be built around each section of lining as it is placed, and all spaces between masonry and linings shall be completely filled with mortar. No broken flue lining shall be used. Flue lining shall start at least 4 in. below the bottom of smoke-pipe intakes of flues, and shall be continued the entire heights of the flues and project at least 4 in. above the chimney top to allow for a 2 in. projection of lining. The wash or splay shall be formed of a rich cement mortar. To improve the draft the wash surface should be concave wherever practical.

Flue lining may be omitted in brick chimneys, provided the walls of the chimneys are not less than 8 in. thick, and that the inner course shall be a refractory clay brick. All brickwork shall be laid in spread mortar, with all joints push-filled. Exposed joints both inside and outside shall be struck smooth. No plaster lining shall be permitted.

Chimneys shall extend at least 3 ft above flat roofs and 2 ft above the ridges of peak roofs when such flat roofs or peaks are within 30 ft of the chimney. The chimney shall be high enough so that the wind from any direction shall not strike the top of the chimney from an angle above the horizontal. The chimney shall be properly capped with stone, terra cotta, concrete, cast-iron, or other approved material; but no such cap or coping shall decrease the flue area.

There shall be but one connection to the flue to which the boiler or furnace smoke-pipe is attached. The boiler or furnace smoke-pipe shall be thoroughly grouted into the chimney and shall not project beyond the inner surface of the flue lining.

The size or area of flue lining or of brick flue for warm-air furnaces depends on height of chimney and capacity of heating system. For chimneys not less than 35 ft in height above grate line, the net internal dimensions of lining should be at least 7 x 11½ in. for a total leader pipe area up to 790 sq in. Above 790 and up to 1,000 sq in. of leader pipe area the lining should be at least 11¼ x 11¼ in. inside. In case of brick flues not less than 35 ft in height with no linings, the internal dimensions should be at least 8 x 12 in. up to 790 sq in. of leader area, and at least 12 x 12 in. for leader capacities up to 1,000 sq in. Chimneys under 35 ft in height are unsatisfactory in operation and hence should be avoided.

CHIMNEYS FOR GAS HEATING

The burning of gas differs from the burning of coal in that the force which supplies the air for combustion of the gas comes largely from the pressure of the gas in the supply pipe, whereas air is supplied to a bed of

burning coal by the force of the chimney draft. If, with a coal-burning boiler, the draft is poor, or if the chimney is stopped, the fire is smothered and the combustion rate reduced. In a gas boiler or furnace such a condition would interfere with the combustion of the gas, but the gas would continue to pass to the burners and the resulting incomplete combustion would produce a dangerous condition. In order to prevent incomplete combustion from insufficient draft, all gas-fired boilers and furnaces should have a back-draft diverter in the flue connection to the chimney.

A study of a typical *back-draft diverter* (Fig. 9) shows that partial or complete chimney stoppage will merely cause some of the products of combustion to be vented out into the boiler room, but will not interfere with combustion. In fact, gas-designed appliances must perform safely

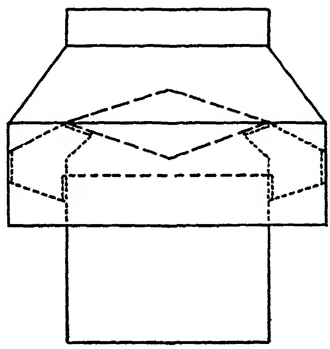


FIG. 9. TYPICAL BACK-DRAFT DIVERTER

under such a condition to be approved by the *American Gas Association* Laboratory. Other functions of the back-draft diverter are to protect the burner and pilot from the effects of down-drafts, and to neutralize the effects of variable chimney drafts, thus maintaining the appliance efficiency at a substantially constant value. Converted boilers or furnaces, as well as gas-designed appliances, should be provided with back-draft diverters.

As is the case with the complete combustion of almost all fuels, the products of combustion for gas are carbon dioxide (CO_2) and water vapor with just a trace of sulphur trioxide (SO_3). Sulphur usually burns to the trioxide in the presence of an iron oxide catalyst. The volume of water vapor in the flue products is about twice the volume of the carbon dioxide when coke oven or natural gas is burned. Because of the large quantity of water vapor which is formed by the burning of gas, it is quite important that all gas-fired central heating plants be connected to a chimney having a good draft. Lack of chimney draft causes stagnation of the products of combustion in the chimney and results in the condensation of a large amount of the water vapor. A good chimney draft draws air into the chimney through the openings in the back-draft diverter, lowers the dew-point of the mixture, and reduces the tendency of the water vapor to condense.

A chimney for a gas-fired boiler or furnace should be constructed in accordance with the principles applicable to other boilers. Where the wall forming a smoke flue is made up of less than an 8 in. thickness of brick, concrete or stone, a burnt fire clay flue tile lining should be used. Care should be used that the lengths of flue tile meet properly with no openings at the joints. Cement mortar should be used for the entire chimney.

TABLE 2. MINIMUM ROUND CHIMNEY DIAMETERS FOR GAS APPLIANCES (INCHES)

Height of Chimney Feet	Gas Consumption in Thousands of Btu per Hour								
	100	200	300	400	500	750	1000	1500	2000
20	4.50	5.70	6.60	7.30	8.00	9.40	10.50	12.35	13.85
40	4.25	5.50	6.40	7.10	7.80	9.15	10.25	12.10	13.55
60	4.10	5.35	6.20	6.90	7.60	8.90	10.00	11.85	13.25
80	4.00	5.20	6.00	6.70	7.35	8.65	9.75	11.50	12.85
100	3.90	5.00	5.90	6.50	7.20	8.40	9.40	11.00	12.40

Table 2 gives the minimum cross-sectional diameters of round chimneys (in inches) for various amounts of heat supplied to the appliance, and for various chimney heights. This is in accordance with *American Gas Association* recommendations.

The flue connections from a gas-fired boiler or furnace to the chimney should be of a non-corrosive material. In localities where the price of gas requires the use of highly efficient appliances, the material used for the flue connection not only should be resistant to the corrosion of water, but should resist the corrosion of dilute solutions of sulphur trioxide in water. Sheet aluminum, as well as some other materials, seems to serve this purpose very well.

Chapter 27

FUELS AND COMBUSTION

Classification of Coal, Air for Combustion, Draft Required, Combustion of Anthracite Coal, Firing Bituminous Coal, Burning Coke, Pulverized Coal, Hand Firing, Classification and Use of Oil, Classification and Use of Gas

COAL, oil, and gas are the principal fuels used for heating. The choice of the fuel to be burned is a question of economy, cleanliness, fuel availability, operation requirements, and control. Information concerning fuel burning devices will be found in Chapter 28.

COAL

The complex composition of coal makes it difficult to classify it into clear-cut types. Its chemical composition is some indication but coals having the same chemical analysis may have distinctly different burning characteristics. Users are mainly interested in the available heat per pound of coal, in the handling and storing properties, and in the burning characteristics. A description of the relationship between the qualities of coals and these characteristics requires considerable space; a treatment applicable to heating boilers is given in *U. S. Bureau of Mines Bulletin* 276.

Fig. 1 illustrates the distribution of the kinds of coal by states and shows the average calorific values and the ash and moisture contents of a limited number purchased on the open market. Five hundred tests in a small domestic boiler were made with these coals by the U. S. Bureau of Mines and the lower curves show the averages of the pounds of water evaporated per pound of coal under similar and normally good operating conditions.

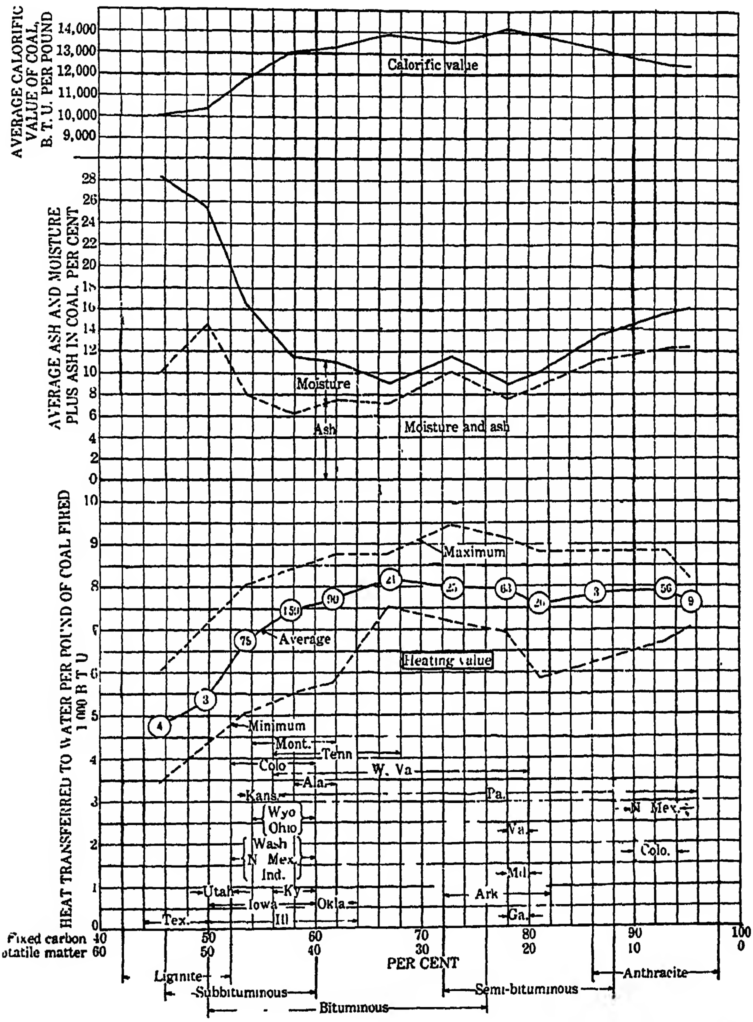
A brief description of the kinds of fuels is given in the following paragraphs, but it should be recognized that there are no distinct lines of demarcation between the kinds, and that they graduate into each other:

Anthracite is a clean, dense, hard coal which creates very little dust in handling. It is comparatively hard to ignite but burns freely when well started. It is non-caking, burns with a short flame, creates very little smoke, burns uniformly, and requires little attention to the fuel beds between firings. It is capable of giving a high efficiency in the common types of hand-fired furnaces.

Semi-anthracite has a higher volatile content than anthracite, is not as hard and ignites somewhat more easily; otherwise its properties are similar to those of anthracite.

Semi-bituminous coal is soft and friable, and fines and dust are created by handling. It ignites somewhat slowly and burns with a medium length of flame. Its caking properties increase as the volatile matter increases, but the coke formed is relatively weak. Having only half the volatile matter content of the more abundant bituminous coals it can be burned with less production of smoke, and is sometimes called *smokeless coal.*

The term *bituminous coal* covers a large range of coals, and includes many types having distinctly different composition, properties and burning characteristics. The coals range from the high-grade bituminous coals of the East to the poorer coals of the West. Their caking properties range from coals which completely melt, to those from which the



Compiled by U. S. Bureau of Mines.

FIG. 1. RELATIVE HEATING VALUES OF COALS WITH CORRESPONDING CALORIFIC VALUES AND PER CENT OF MOISTURE AND ASH

volatiles and tars are distilled without change of form, so that they are classed as non-caking or free-burning. Most bituminous coals are strong and non-friable enough to permit of the screened sizes being delivered free from fines. In general, they ignite easily, burn freely; the length of flame varies with different coals, but it is long. Much smoke and soot are possible and are difficult to reduce to reasonable amounts at low rates of burning.

Sub-bituminous coals occur in the western states; they are high in moisture when

mined and tend to break up as they dry or when exposed to the weather; they are liable to ignite spontaneously when piled or stored. They ignite easily and quickly and have a medium length flame, are non-caking and free-burning; the lumps tend to break into small pieces if poked; very little smoke and soot are formed.

Lignite is of woody structure, very high in moisture as mined, and of low heating value; it is clean to handle. It has a greater tendency than the sub-bituminous coals to disintegrate as it dries, and also is more liable to spontaneous ignition. Freshly mined lignite, because of its high moisture, ignites slowly. It is non-caking. The char left after the moisture and volatile matter are driven off burns very easily, like charcoal. The lumps tend to break up in the fuel bed and pieces of char falling into the ash pit continue to burn. Very little smoke or soot is formed.

Coke is produced by the distillation of the volatile matter from coal. The type of coke depends on the coal, or mixture of coals used, the temperatures and time of distillation and, to some extent, on the type of retort or oven; coke is also produced as a residue from the destructive distillation of oil.

High-temperature cokes. Coke as usually available is of the high-temperature type, and contains between 1 and 2 per cent volatile matter. High-temperature cokes are subdivided into *beehive coke* of which comparatively little is now sold for domestic use, *by-product coke,* which covers the greater part of the coke sold, and *gas-house coke.* The differences among these three cokes are relatively small; their denseness and hardness decrease and friability increases in the order named. In general, the lighter and more friable cokes ignite and burn the more easily.

Low-temperature cokes are produced at lower coking temperatures, and only a portion of the volatile matter is distilled off. Cokes as made by various processes under development have contained from 10 to 15 per cent volatile matter. In general, these cokes ignite and burn more readily than high-temperature cokes. The properties of various low-temperature cokes may differ more than various high-temperature cokes because of the differences in the quantities of volatile matter and because some may be light and others are briquetted.

The sale of *petroleum coke* for domestic furnaces has been small and is generally confined to the Middle West. They vary in the amount of volatile matter they contain, but all have the common property of a very low ash content, which necessitates the use of refractory pieces to protect the grates from being burned.

In order to obtain perfect combustion a definite amount of air is required for each pound of fuel fired. A deficiency of air supply will result in combustible products passing to the stack unburned. An excess of air absorbs heat from the products of combustion and results in a greater loss of sensible heat to the stack.

Total Air Required. The theoretical amount of air required per pound of fuel for perfect combustion is dependent upon the analysis of the fuel;

TABLE 1. POUNDS OF AIR PER POUND OF FUEL AS FIRED

ANTHRACITE	COKE	SEMI-BITUMINOUS	BITUMINOUS	LIGNITE
9.6	11.2	11.2	10.3	6.2

however, for estimating purposes the theoretical air required for different grades of fuel may roughly be taken from Table 1. An excess of about 50 per cent over the theoretical amount is considered good practice under usual operating conditions.

The amount of excess air, based upon the laws of combustion, can be determined by its relation to the percentage of CO_2 (carbon dioxide) in the products of combustion. This relationship is shown by the curves (Fig. 2) for high and low volatile coals and for coke. In hand-fired furnaces with long periods between firings the combustion goes through a cycle in each period and the quantity of excess air present varies.

Secondary Air. The division of the total into primary and secondary air necessary to produce the same rate of burning and the same excess air depends on a number of factors which include size of fuel, depth of fuel bed and diameter of fire pot. The ratio of the secondary to the primary air increases with decrease in the size of the fuel pieces, with increase in the depth of the fuel bed, and with increase in the area of the fire pot; the ratio also increases with increase in rate of burning.

Size of the fuel is a very important factor in fixing the quantity of secondary air required for non-caking coals. With caking coals it is not so important because small pieces fuse together and form large lumps. Fortunately a smaller size fuel gives more resistance to air flow through the fuel bed and thus automatically causes a larger draft above the fuel

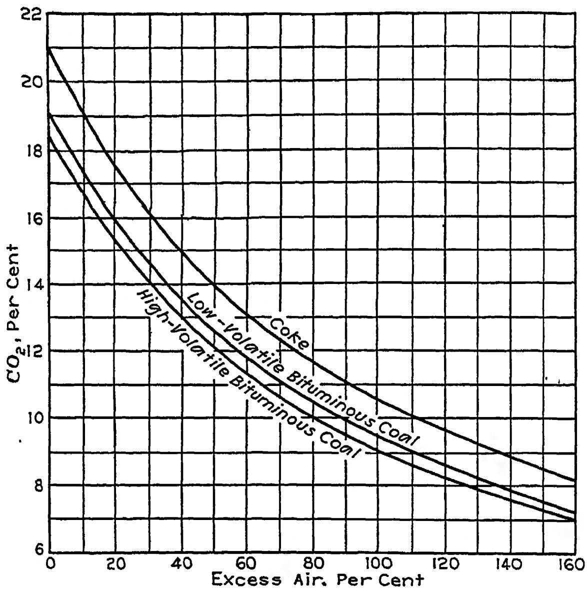


FIG. 2. RELATION BETWEEN CO_2 AND EXCESS AIR IN GASES OF COMBUSTION

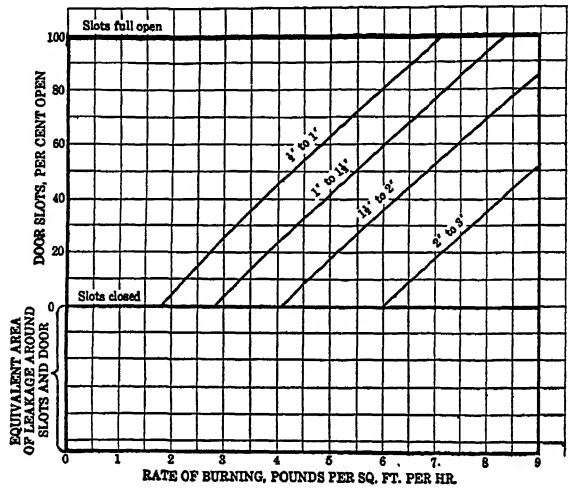
bed, which draws in more secondary air for the same slot openings. In spite of this, a small size fuel requires a larger opening of the door slots; for a certain size for each fuel no slot opening is required, and for larger sizes too much excess air gets through the fuel bed.

It is impossible to establish a single rule for the correct slot opening for all types and sizes of fuels and for all rates of burning. Furthermore, the size of slot opening is dependent on whether the ashpit damper is open or closed. It is better to have too much than too little secondary air; the opening is too small if there is a puff of flame when the firing door is opened.

Fig. 3 taken from the *U. S. Bureau of Mines Report of Investigations* No. 2980 shows the relationship of the slot opening, for a domestic furnace, to the size of coke and the rate of burning; these openings are with the ashpit damper wide open, and would be less if the available draft permits of its being partly closed. The same openings are satisfactory for anthracite.

Bituminous coals require a large amount of secondary air during the period subsequent to a firing, to consume the gases and to reduce the smoke. The smoke produced is a good indicator, and that opening is best which reduces the smoke to a minimum. Too much secondary air will cool the gases below the ignition point, and prove harmful instead of beneficial. The following suggestions will be helpful:

1. In cold weather, with high combustion rates, the secondary air damper should be half open all the time.
2. In very mild weather, with a very low combustion rate, the secondary air damper should be closed all the time.



From U. S. Bureau of Mines.

Fig. 3. Relative Amount of Fire Door Slot Opening Required in a Given Furnace to Give Equally Good Combustion for High Temperature Coke of Various Sizes When Burned at Various Rates

3. For temperatures between very mild and very cold, the secondary air damper should be in an intermediate position.
4. For ordinary house operation, secondary air is needed after each firing for about one hour.

Draft Required

The draft required to effect a given rate of burning the fuel as measured at the smokehood is dependent on the following factors:

1. Kind and size of fuel.
2. Combustion rate per square foot of grate area per hour.
3. Thickness of fuel bed.
4. Type and amount of ash and clinker accumulation.
5. Amount of excess air present in the gases.
6. Resistance offered by the boiler passes to the flow of the gases.
7. Accumulation of soot in the passes.

Insufficient draft will necessitate additional manipulation of the fuel

bed and more frequent cleanings to keep its resistance down. Insufficient draft also restricts the control by adjustment of the dampers.

The quantity of excess air present has a marked effect on the draft required to produce a given rate of burning, and it is often possible to produce a higher rate by increasing the thickness of the fuel bed.

Combustion of Anthracite Coal

An anthracite coal fire should never be poked, as this serves to bring ash to the surface of the fuel bed where it melts into clinker.

Grate size anthracite coal is suitable for large grates and for a fuel depth of 20 in. or more. The air spaces are large; the amount of surface in contact with the air is small, and consequently combustion is slow. Lumps of rock or slate, being large, may cause trouble in shaking the grate. On grates 25 in. and under, the fire is likely to go out easily with this fuel, unless mixed with small size coal for reducing the air spaces.

Egg size is suitable for large firepots (grates 25 in. and over) if the fuel can be fired at least 20 in. deep. The air spaces between the pieces of coal are large and normal combustion is slower than with the next smaller size coal. Pieces of rock and slate sometimes may be large enough to cause some trouble in shaking the grate. For best results this coal should be fired deep.

Stove size coal is the popular size of anthracite fuel for most boilers and furnaces used for heating buildings. It is small enough to burn well on 17 in. grates, but large enough so that the draft loss through the fuel bed is not too great. The only instructions needed for burning this type of fuel are that the grate should be shaken before each firing, the fire should never be poked from the top, and the fuel should be fired deeply and uniformly.

Chestnut size coal is in demand for firepots up to 20 in. in diameter, especially those in which the fuel cannot be fired over 15 in. deep. The percentage of ash is usually higher than in stove coal, but the pieces of rock and slate are small and do not interfere with the shaking of the grates.

Pea size coal is often an economical fuel to burn. It is relatively low in price. When fired carefully, pea coal can be burned on standard grates. It is well to have a small amount of a larger fuel on hand when building new fires, or when filling holes in the fuel bed. Care should be taken to shake the grates only until the first bright coals begin to fall through the grates. The fuel bed, after a new fire has been built, should be increased in thickness by the addition of small charges until it is at least level with the sill of the fire door. This keeps a bed of ignited coal in readiness against the time when a sudden demand for heat shall be made on the heater. When firing, the bright fuel should either be pulled forward or pushed to the back of the firepot, leaving a hollow in which to throw the fresh fuel and leaving a portion of the glowing coal exposed to ignite the gases rising from the fuel; otherwise a gas explosion may occur.

Pea size coal requires a strong draft and therefore the best results generally will be obtained by keeping the choke damper open, the cold-air check closed, and by controlling the fire with the air-inlet damper only. Pea size can also be fired in layers with stove or egg size anthracite and its use in this manner will reduce the fuel costs and attention required.

Buckwheat size coal requires much the same attention as pea size coal, except that the smaller size of the fuel makes it more difficult to burn on ordinary grates. Even greater care must be taken in shaking the grates than with pea coal on account of the danger of the fuel falling through the grate. A good draft is required and consequently the fire is best controlled by the air-inlet damper only. Where frequent attention can be given and where there is not a big heat demand, this fuel is frequently burned without the aid of any special equipment.

In general it will be found more satisfactory with buckwheat coal to maintain a uniform heat output and consequently to keep the system warm all the time, rather than to allow the system to cool off at times and then to attempt to burn the fuel at a high rate while warming up. A uniform low fire will minimize the clinker formation and keep the clinker in an easily broken up condition so that it readily can be shaken through the grate.

Forced draft and special grates or retorts frequently are used with this fuel for best results.

No. 2 buckwheat anthracite, or rice size, is used only with forced draft equipment on mechanical stokers. No. 3 buckwheat anthracite, or barley, has no application in domestic heating.

Firing Bituminous Coal

Bituminous coal should never be fired over the entire fuel bed at one time. A portion of the glowing fuel should always be left exposed to ignite the gases leaving the fresh charge.

Air should be admitted over the fire through a special secondary air device, or through a slide in the fire door or by opening the fire door slightly. If the quantity of air admitted is too great the gases will be cooled below the ignition temperature and will fail to burn. The fireman can judge the quantity of air to admit by noting when the air supplied is just sufficient to make the gases burn rapidly and smokelessly above the fuel bed.

The red fuel in the firebox, before firing, excepting only a shallow layer of coke on the grate, should be pushed to one side or forward or backward to form a hollow in which to throw the fresh fuel. (Some manufacturers recommend that all red fuel be pushed to the rear of the firebox and that the fresh fuel be fired directly on the grate and allowed to ignite from the top. The object of this is to reduce the early rapid distillation of gases and to reduce the quantity of secondary air required for smokeless combustion).

It is well to have the bright fuel in the firebox so placed that the gases from the freshly fired fuel, mixed with the air over the fuel bed, pass over the bed of bright fuel on the way to the flues. The bed of bright fuel then supplies the heat to raise the mixture of air and gas to the ignition temperature, thereby causing the gaseous matter to burn and preventing the formation of smoke.

The fuel bed should be carried as deep as the size of fuel and the available draft permit, in order to have as much coked fuel as possible for pushing to the rear of the firebox at the time of firing. A deep fuel bed obtains the longest firing intervals.

If the coal is of the caking kind the fresh charge will fuse into one solid mass which can be broken up with the stoking bar and leveled from 20 min to one hour after firing, depending on the temperature of the firebox. Care should be exercised when stoking not to bring the bar up to the surface of the fuel as this will tend to bring ash into the high temperature zone at the top of the fire, where it will melt and form clinker. The stoking bar should be kept as near the grate as possible and should be raised only enough to break up the fuel. With fuels requiring stoking it may not be necessary to shake the grates, as the ash is usually dislodged during stoking.

The output obtained from any heater with bituminous coal will usually exceed that obtainable with anthracite, since soft coal burns more rapidly than hard coal and with less draft. Soft coal, however, will require frequent attention to the fuel bed, because it burns unevenly, even though the fuel bed may be level, forming holes in the fire which admit too much air, chilling the gases over the fuel bed and reducing the available draft.

Semi-bituminous coal is fired like bituminous coal, and because of its caking characteristics requires practically the same attention. The *Pocahontas Operators Association* recommends the central cone method of firing, in which the coal is heaped on to the center of the bed forming a cone the top of which should be level with the middle of the firing door. This allows the larger lumps to fall to the sides, and the fines to remain in the center and be coked. The poking should be limited to breaking down the coke without stirring and to gently rocking the grates. It is recommended that the slides in the firing door be kept closed, as the thinner fuel bed around the sides allows enough air to get through.

Burning Coke

Coke is a very desirable fuel and usually will give satisfaction as soon as the user learns how to control the fire. Coke ignites and burns very rapidly with less draft than anthracite coal. In order to control the air admitted to the fuel it is very important that all openings or leaks into the ashpit be closed tightly. A coke fire responds more rapidly than an anthracite fire to the opening of the dampers. This is an advantage in warming up the system, but it also makes it necessary to watch the dampers more closely in order to prevent the fire from burning too rapidly. A deep fuel bed always should be maintained when burning coke. The grates should be shaken only slightly in mild weather and should only be shaken until the first red particles drop from the grates in cold weather. Since coke weighs only about half as much as anthracite coal per cubic foot, and therefore only about half as much can be put in the firepot, it will be necessary to fire oftener; but during the greater part of the heating season this will not be an item of importance. The best size of coke for general use, for small firepots where the fuel depth is not over 20 in., is that which passes over a 1 in. screen and through a 1½ in. screen. For large firepots where the fuel can be fired over 20 in. deep, coke which passes over a 1 in. screen and through a 3 in. screen can be used, but a uniform size of coke is always more satisfactory. Large sizes of coke should be either mixed with fine sizes or should be broken up before using.

The practice of treating the more friable coals to allay the dust they create is increasing. The coal is sprayed with a solution of calcium chloride or a mixture of calcium and magnesium chlorides. Both these salts are very hygroscopic and their moisture under normal atmospheric conditions keeps the surface of the coal damp, thus reducing the dust during delivery and in the cellar, and obviating the necessity of sprinkling the coal in the bin.

The coal is sometimes treated at the mine, but more usually by the local distributor just before delivery. The solution is sprayed under high pressure, using from 2 to 4 gal or from 5 to 10 lb of the salt per ton of coal, depending on its friability and size.

Installations of pulverized coal burning plants in heating boilers are of the unit type, in which the pulverized coal is delivered into the furnace immediately after grinding, together with the proper amount of preheated air. With this apparatus, where the necessary furnace volume is obtainable, high efficiencies can be obtained.

A 150-hp boiler has generally been considered the smallest size for which pulverized fuel is feasible. Complications are introduced if an installation with a single boiler has to take care of very light loads.

Hand Firing

Hand firing is the oldest and the most widely used method of burning coal for heating purposes. To keep the fuel bed in proper condition where hand firing is used, the following general rules should be observed:

1. Remove ash from fuel bed by shaking the grates whenever fresh fuel is fired. This removes ash from the fire, enables the air to reach the fuel and does away with the formation of clinker (which is melted ash).
2. Supply the boiler with a deep bed of fuel. Nothing is gained by attempting to fire a small amount of fuel. A deep bed of fuel secures the most economical results.
3. Remove ash from ashpit at least once daily. Never allow ash to accumulate up to the grates. If the ash prevents the air from passing through, the grate bars will burn out and much clinker trouble will be experienced.

The principal requirements for a *hand-fired furnace* are that it shall have enough grate area and combustion space. The amount of grate area required is dependent upon the desired combustion rate.

The furnace volume is influenced by the kind of coal used. Bituminous coals, on account of their long-flaming characteristic, require more space in which to burn the gases of combustion completely than do the coals low in volatile matter. For burning high volatile coals provision should be made for mixing the combustible gases thoroughly so that combustion is complete before the gases come in contact with the relatively cool heating surfaces. An abrupt change in the direction of flow tends to mix the gases of combustion more thoroughly.

OIL

Uniform oil specifications were prepared in 1929 by the *American Oil Burner Association*, in cooperation with the *American Petroleum Institute*, the U. S. Bureau of Standards, the *American Society for Testing Materials* and other interested organizations. Oil fuels were classified into six groups, as indicated by Table 2. When these specifications were prepared, it was generally accepted that the first three grades were adapted to domestic

manufactured gas. Most states have legislation which controls the distribution of gas and fixes a minimum limit to its heat content. The gross or higher calorific value usually ranges between 520 and 545 Btu per cubic foot, with an average of 535. A given heat value may be maintained and yet leave considerable latitude in the composition of the gas so that as distributed the composition is not necessarily the same in different districts, nor at successive times in the same district. There are limits to the variation allowable, because the specific gravity of the gas depends on its

TABLE 3. REPRESENTATIVE PROPERTIES OF GASEOUS FUELS, BASED ON GAS AT 60 F AND 30 IN. HG..

GAS	BTU PER CU FT		SPECIFIC GRAVITY, AIR = 1.00	AIR REQUIRED FOR COMBUSTION, (CU FT)	PRODUCTS OF COMBUSTION				THEORETICAL FLAME TEMPERATURE, (DEG FAHR)
					Cubic Feet			ULTIMATE CO_2 Dry Basis	
	High (Gross)	Low (Net)			CO_2	H_2O	Total with N_2		
Natural gas—Mid-Continental	967	873	0.57	9.17	0.97	1.92	10.2	11.7	3580
Natural gas—Ohio	1130	1025	0.65	10.70	1.17	2.16	11.8	12.1	3600
Natural gas—Pennsylvania	1232	1120	0.71	11.70	1.30	2.29	12.9	12.3	3620
Retort coal gas	575	510	0.42	5.00	0.50	1.21	5.7	11.2	3665
Coke oven gas	588	521	0.42	5.19	0.51	1.25	5.9	11.0	3660
Carburetted water gas	536	496	0.65	4.37	0.74	0.75	5.0	17.2	3815
Blue water gas	308	281	0.53	2.26	0.46	0.51	2.8	22.3	3800
Anthracite producer gas	134	124	0.85	1.05	0.33	0.19	1.9	19.0	3000
Bituminous producer gas	150	140	0.86	1.24	0.35	0.19	2.0	19.0	3160
Oil gas	575	510	0.35	4.91	0.47	1.21	5.6	10.7	3725

composition, and too great a change in the specific gravity necessitates a change in the adjustment of the burners of small appliances.

Table 3 shows that a large proportion of the products of combustion when gas is burned, may consist of water vapor, and that the greater the proportion of water vapor, the lower the maximum attainable CO_2 by gas analysis. The table also shows that a low calorific value does not necessarily mean a low flame temperature since, for example, natural gas has a theoretical flame temperature of 3600 F and blue water gas of 3800 F, although it has a calorific value less than one-third that of natural gas.

The quantity of air given in Table 3 is that required for theoretical combustion, but with a properly designed and installed burner, the excess air can be kept low. The division of the air into primary and secondary is a matter of burner design and the pressure of gas available, and also of the type of flame desired.

Chapter 28

AUTOMATIC FUEL BURNING EQUIPMENT

Stokers, Residential Stokers, Apartment House Stokers, Commercial Stokers, Domestic Oil Burners, Air and Oil Supply, Atomization, Type of Flame, Ignition, Boilers for Domestic Oil Burners, Commercial Oil Burners, Gas-Fired Appliances, Gas Boilers, Warm Air Furnaces, Space Heaters, Conversion Burners, Gas Appliances, Installation Features

LABOR saving, automatic, mechanical equipment for the efficient combustion of coal, oil, and gas is considered in this chapter.

MECHANICAL STOKERS

Assuming the same intelligence in handling the fire, coal can be burned more efficiently on a mechanical stoker than on any kind of hand-fired grate. This does not necessarily mean that a stoker installation may be more economical, because the amount of coal burned may be so small or the cost of the installation so high that the savings with stokers may not be sufficient to pay for the investment. The operation of burning coal involves uniformity in stoking, proper distribution over the fuel bed, admission of air as required to all parts of the fuel bed, and disposal of the ash. The handling of the volatile gas is largely a matter of furnace design but since this gas forms a considerable portion of the heating value of the coal, it may also be said that the proper handling of this gas is a function of firing. All mechanical stokers must provide means of taking care of these several functions in order fully to serve their purpose.

Classifications of Stokers

Stokers may be divided into four types according to their construction and operation, namely, (1) overfeed flat grate, (2) overfeed inclined grate, (3) underfeed side cleaning type, and (4) underfeed rear cleaning type. They may also be classified according to their uses. The following classification has been adopted by the U. S. Department of Commerce:

Class 1. Residential (Capacity less than 100 lb coal per hour).

Class 2. Apartment houses and small commercial heating jobs (Capacity 100 to 200 lb coal per hour).

Class 3. General commerical heating and small high pressure steam plants (Capacity 200 to 300 lb coal per hour).

Class 4. Large commercial and high pressure steam plants (Capacity over 300 lb per hour).

Overfeed Flat Grate Stokers

This type is represented by the various chain grate stokers. These

stokers receive fuel at the front of the grate in a layer of uniform thickness and move it back horizontally to the rear of the furnace. Air is supplied under the moving grate to carry on combustion at a sufficient rate to complete the burning of the coal near the rear of the furnace. The ash is carried over the back end of the stoker into an ashpit beneath. This type of stoker is suitable for small sizes of anthracite or coke breeze and also for bituminous coals the clinker forming characteristics of which make it desirable to burn the fuel without disturbing it. This type of stoker invariably requires the use of an arch over the front of the stoker to maintain ignition of the incoming fuel and to maintain the volatile gas at a temperature suitable for combustion. Frequently, a rear combustion arch is required to maintain ignition until the fuel is fully consumed.

Overfeed Inclined Grate Stokers

In general the combustion principle is similar to the flat grate stoker, but this stoker is provided with rocking grates set on an incline to advance the fuel during combustion. Also this type is provided with an ash plate where ash is accumulated and from which it is dumped periodically. This type of stoker is suitable for all types of coking fuels but preferably for those of low volatile content. Its grate action has the tendency to keep the fuel bed well broken up thereby allowing for free passage of air. Because of its agitating effect on the fuel it is not so desirable for badly clinkering coals. Furthermore, it should usually be provided with a front arch to care for the volatile gas.

Underfeed Side Cleaning Stokers

In this type, the fuel is fed in at the front of the furnace to one or more retorts, is advanced away from the retort as combustion progresses, while finally the ash is disposed of at the sides. This type of stoker is suitable for all coking coals while in the smaller sizes it is suitable for small sizes of anthracites. In this type of stoker the fuel is delivered to a retort beneath the fire and is raised into the fire. During this process the volatile gas is released, is mixed with air, and passes through the fire where it is burned. The ash may be continuously discharged as in the small stoker or may be accumulated on a dump plate and periodically discharged. This stoker requires no arch as it automatically provides for the combustion of the volatile gas.

Underfeed Rear Cleaning Stokers

This type carries on combustion in much the same manner as the side cleaning type, but consists of several retorts placed side by side and filling up the furnace width, while the ash disposal is at the rear. In principle, its operation is the same as the side cleaning underfeed.

Class 1 Stokers, Residential

A common type of stoker in this class consists of a round retort having tuyeres at the top where all of the air for combustion is admitted. Coal is fed from a storage hopper outside of the boiler by means of a worm into the bottom of this retort and beneath the fire. The equipment includes a blower which is driven by the same motor that drives the stoker.

Some domestic stokers are provided with automatic grate shaking

mechanism together with screw conveyers for removing the ash from the ashpit and depositing it in an ash receptacle outside the boiler. Certain types can also be provided with a coal conveyer which takes coal from the storage bin and maintains a full hopper at the stoker. They may feed coal to the furnace either intermittently or with a continuous flow regulated automatically to suit conditions. Where the boiler is provided with indirect coils for heating the domestic hot water, the stoker may be so arranged that it can be used the entire year to maintain a continuous hot water supply.

Class 2 Stokers, Apartment House

This class is used extensively for heating plants in apartments, hotels, etc., and also for small industrial plants such as laundries, bakeries, creameries, etc. The various stokers in this class differ materially in their design, although the majority are of the underfeed type. The principal exception is an overfeed type having step action grates in a horizontal plane and so arranged that they are alternately moving and stationary, and are designed to advance the fuel during combustion to an ash plate at the rear.

All of the stokers are provided with a coal hopper outside of the boiler. In the underfeed types, the coal feed from this hopper to the furnace may be accomplished by a continuously revolving worm or by an intermittent plunger. The drive for the coal feed may be an electric motor, or a steam or hydraulic cylinder. With an electric motor, the connection between the driver and the coal feed may be through a variable speed gear train which provides two or more speeds for the coal feed; or it may be through a simple gear train and a variable speed driver for the change in speed of the coal feed; or a simple gear train with a coal feed having an adjustment for varying the travel of the feeding device. With a steam or hydraulic cylinder, the power piston is connected directly to the coal feeding plunger.

The stokers in this class vary also in their retort design. It is customary in the worm-feed type to use a short retort in order that the unsupported length of worm within the retort may not be too weak for continuous service. In this type the retort is placed approximately in the middle of the furnace and is provided with tuyere openings at the top on all sides. In the plunger-feed type the retort extends from the inside of the front wall entirely to the rear wall or to within a short distance of the rear wall. This type of retort has tuyeres on the sides and at the rear.

This class of stokers also differs in the grate surface surrounding the retort. In many of the worm-feed stokers this grate is entirely a dead plate on which the fuel rests while combustion is completed. In the dead-plate type, all of the air for combustion is furnished by the tuyeres at the retort. Because of this, combustion is well advanced over the retort so that it may easily be completed by the air which percolates through the fuel bed. With the dead-plate type of grate the ash is removed through the fire doors and it is therefore desirable that the fuel used shall be one in which the ash is readily reduced to a clinker at the furnace temperature, in order that it may be removed with the least disturbance of the fuel bed.

In other stokers in this class, the grates outside of the retort are air-

admitting and shaking grates. These grates permit a large part of the ash to be shaken into the ash pit beneath, while the clinkers are removed through the fire doors. With this type of grate, the main air chamber extends only under the retort while the side grates receive air by natural draft from the ash pit.

In still other stokers of this class, the main air chamber extends beyond the retort and is covered with fuel-bearing, air-supplying grates. With this type of grate, the fuel is supplied with air from the main air chamber throughout combustion. Also with this type of grate, dump plates are provided beyond the grates where the ash accumulates and from which it can be dropped periodically into the ash pit beneath.

Stokers in this class are compactly built in order that they may fit into standard heating boilers and still leave room for sufficient combustion space above the grates. The height of the grate is approximately the same as that of the ordinary grates of boilers, so that it is usually possible to install such stokers with but minor changes in the existing equipment. In some districts, there are statutory regulations governing such settings.

These stokers vary in furnace dimensions from 30 in. square to approximately 66 in. square. The capacity of the stokers is measured by the amount of coal that can be burned per hour. In general, manufacturers recommend that, for continuous operation, the coal burning rate shall not exceed 25 lb of coal per square foot of grate per hour, while for short peaks this rate may be increased to 30 lb per hour. Although these stokers were designed to burn bituminous coal, they can also be used to burn the small sizes of anthracite but at a somewhat lower rate. It is often customary to have the janitor or some other attendant care for the boiler as one of his duties. Under these conditions the heating plant does not receive the same careful attention as it would if a man devoted his entire attention to the fire. With periodic hand-firing, the boiler is operated inefficiently much of the time. With a stoker, the boiler is operated at the rate that the conditions require so long as there is coal in the hopper. With hand firing, it is customary to use the more expensive sizes of fuel, while with a stoker the smaller sizes are used at a considerable saving in the cost per ton. Because the stoker responds promptly to automatic regulation, it is possible to maintain a reasonably constant standard. Also because of the fact that the stoker feeds the fuel regularly and in small quantities without losses due to opening doors, etc., it must of necessity be more efficient than hand firing. This increase in efficiency depends entirely on conditions, with a minimum of about 10 per cent and a maximum of about 25 per cent.

Class 3 Stokers, General Commercial

These stokers are suitable for the heating plants of large schools, hotels, hospitals, or other large institutions as well as industrial plants. This class is served both by overfeed stokers and by underfeed stokers. The overfeed stokers are in general of three types, (1) the chain grate, (2) the rear cleaning inclined grate, and (3) the center cleaning inclined grate or V type.

Stokers of this type are usually operated by natural draft, although in some cases conditions permit the operation of forced draft under the grates. With most fuels, it is not advisable to operate overfeed grates at

too high a combustion rate because of the greater difficulty of cleaning and the higher maintenance, but where the fuel is free burning and has a high ash fusion temperature, the combustion rate is not so restricted. The operation of the chain grates and the rear cleaning type of inclined grates has already been described.

The V-type stoker is practically obsolete although many are still in operation. In this stoker, the grates are inclined downward from both sides of the furnace to a low point at the middle where there is either a dump plate for periodic disposal of the ash or a rotary ash grate for continuous discharge of ash. In this stoker, the fuel is fed into a hopper at the top of the grate on each side of the furnace and advanced down the grates to the center where the refuse is accumulated. This stoker is always provided with a combustion arch over the entire furnace for the purpose of assuring thorough combustion of the solid fuel and providing a furnace temperature sufficiently high to burn the volatile gases. Because of this high furnace temperature and because so little of the boiler surface is exposed to the fire to assist in carrying off the heat by radiation, this stoker is characterized by severe clinkering in the ash area. With all types of overfeed stokers, the most desirable installations are in boilers which are operated with comparatively uniform loads and moderate rates of combustion, since, even with good combustion arches, fluctuating loads or high combustion rates result in free volatile gas and this in turn means smoke.

The underfeed stokers in this class were the first of the type to be developed as at the time of their development very few large boilers were in use. The stokers are not so varied in design as those in the smaller class although in principle they are much the same. Practically all of them are of the plunger coal feed type with retorts extending the entire length of the furnace, with air supplying grates adjacent to the retorts, and with manually-operated dump plates at the sides of the furnace. The coal feeding plunger is operated by a steam or electric driver through a reduction gearing, or by a steam or hydraulic piston connected directly to the coal feeding plunger.

These stokers are heavily built and designed to operate continuously at high boiler ratings with a minimum amount of attention. Because of the fact that all volatile gas must pass through the fire before reaching the combustion chamber, these stokers will operate smokelessly under ordinary conditions. Also because of the fact that these stokers are always provided with forced draft, they are the most desirable type for fluctuating loads or high boiler ratings.

In the design of the grates for supporting the fuel between the retort and the ash plates, the stokers differ in providing for movement of the fuel during combustion. Some stokers are designed with fixed grates of sufficient angle to provide for this movement as the bed is agitated by the incoming fuel, while others have alternate moving and stationary bars in this area and provide for this movement mechanically. In either type, with proper operation, all refuse will be deposited at the dump plate. Another difference in these stokers is that some makes use a single air chamber under the whole grate area thus having the same air pressure under the ignition area as under the rest of the grate, while others have a divided air chamber using the full air pressure under the ignition area and

a reduced air pressure under the remainder of the grate. These stokers vary in size from approximately 5 ft square to a maximum of 8½ ft square.

Class 4 Stokers, Large Commercial

These stokers are usually of the underfeed type with multiple retorts and either side cleaning or rear cleaning. In the side cleaning type there may be as many as three retorts in the furnace, and the stoker functions in the same manner as has been described for the single retort. These stokers are usually limited in length to approximately 8½ ft while the width may be as great as 10½ ft. In the rear cleaning stokers the number of retorts and the dimensions of the furnace are practically unlimited.

DOMESTIC OIL BURNERS

The number of combinations of the characteristic elements of domestic oil burners is rather large and accounts for the variety of burners found in actual practice. Domestic oil burners may be classified as follows:

1. AIR SUPPLY FOR COMBUSTION
   - *a. Atmospheric*—by natural chimney draft.
   - *b. Mechanical*—electric-motor-driven fan or blower.
   - *c. Combination of (a) and (b)*—primary air supply by fan or blower and secondary air supply by natural chimney draft.
2. METHOD OF OIL PREPARATION
   - *a. Vaporizing*—oil distills on hot surface or in hot cracking chamber.
   - *b. Atomizing*—oil broken up into minute globules.
     - (1) Centrifugal—by means of rotating cup or disc.
     - (2) Pressure—by means of forcing oil under pressure through a small nozzle or orifice.
     - (3) Air or steam—by high velocity air or steam jet in a special type of nozzle.
     - (4) Combination air and pressure—by air entrained with oil under pressure and forced through a nozzle.
   - *c. Combination of (a) and (b).*
3. TYPE OF FLAME
   - *a. Luminous*—a relatively bright flame. An orange-colored flame is usually best if no smoke is present.
   - *b. Non-luminous*—Bunsen-type flame (*i.e.* blue flame).
4. METHODS OF IGNITION
   - *a. Electric.*
     - (1) Spark—by transformer producing high-voltage sparks. Usually shielded to avoid radio interference. May take place continuously while the burner is operating or just at the beginning of operation.
     - (2) Resistance—by means of hot wires or plates.
   - *b. Gas.*
     - (1) Continuous—pilot light of constant size.
     - (2) Expanding—size of pilot light expanded temporarily at the beginning of burner operation.
   - *c. Combination*—electric sparks light the gas and the gas flame ignites the oil.
   - *d. Manual*—by manually-operated gas torch for continuously operating burners.

5. MANNER OF OPERATION
   a. *On and off*—burner operates only a portion of the time (intermittent).
   b. *High and low*—burner operates continuously but varies from a high to a low flame.
   c. *Graduated*—burner operates continuously but flame is graduated according to needs by regulating both air and oil supply.

Air and Oil Supply

The object of regulating the air and oil supplies is to obtain a complete mixture of the proper quantities of oil and air so the fire will be clean and efficient. Proper and dependable ignition also depends upon the ability of the burner to produce consistent fuel-air mixtures. The type and shape of flame depend largely upon the methods of air and oil supply employed.

It should be pointed out that this mixture burns in a space called the furnace, which is lined with refractory bricks or other heat-resistant substances for the purpose of maintaining that space at a high temperature so that the oil and air may completely unite and burn. Excessive cooling before combustion is completed stops the combustion process and causes soot. The furnace is in some instances a valuable auxiliary in assisting in the actual mixture of oil and air and in modifying the flame shape, besides its primary function of maintaining high temperatures. The size and shape of furnace required are important, especially where the dimensions of the space into which the burner is to be placed are already fixed.

Atomization

The purpose of atomization is greatly to increase the surface area of a given quantity of oil in order to accelerate the change from the liquid state (in which oil cannot burn) to the gaseous or vaporous state, in which state it is one of the elementary fuels, gaseous hydrocarbon. This conversion is largely accomplished through the action of radiant heat energy upon the flying globules of oil, and the tremendously increased surface provided aids gasification.

Air for Combustion

Air for combustion usually is supplied by a motor-driven fan, several types being in common use. Electric motors varying from $\frac{1}{20}$ hp to $\frac{1}{8}$ hp are used and are started and stopped by the control mechanism. In most cases, they are direct-coupled to the fan as well as to a gear or lobe pump for drawing the oil from the storage tank, and in some cases, to a pump for forcing the oil through the nozzle.

All of the air required for combustion can be supplied by the blower, or else only the *primary* air can be supplied under pressure and provision made so that the remainder will be drawn into the combustion chamber by the natural draft developed by the chimney or by an injector-like action of the primary air. In any event there should be definite control of the quantity of air as well as of the rate of oil supply. Some method of draft regulation is advisable in order to secure proper air regulation. It is necessary to supply more air than is actually required for complete combustion of the oil, but the amount of excess air should be reduced to the

lowest workable minimum. Laboratory tests frequently show 25 per cent to 50 per cent more air than is required for combustion, yet field tests indicate that the average burner operates with from 50 per cent to 125 per cent excess air, with a corresponding reduction in the efficiency of the burner. Many domestic burners are extremes of simplicity, the only moving parts being the motor armature with a shaft and direct-connected fan and pump set.

Type of Flame, Ignition

If the vaporizing of the atomized oil and the combustion are concurrent events, a luminous flame will usually result. If vaporization and mixing are accomplished before combustion, a non-luminous flame will result. Some burners may produce either type of flame according to the adjustment made.

A limited comparison of these two types of flame shows no inherent superiority of one over the other so far as thermal efficiencies are concerned. This is definitely true when the burners are placed in boilers having ample indirect surface, and is probably true in general. The moot question of radiation has not been conclusively settled. There are indications that the radiation of luminous and non-luminous flames in boiler furnaces are practically the same. More information is needed upon this subject.

It is true, however, that a non-luminous flame may show low excess air and the presence of carbon monoxide, but no smoke. Low excess air with a luminous flame will usually show little or no carbon monoxide, but will be unmistakably smoky. Visual indications, especially with a blue flame, may therefore be quite unreliable.

When a burner is operating intermittently under the control of a thermostat, some positive form of ignition is required to function every time there is a call for heat.

The necessity for certain ignition under adverse conditions, when the line voltage is low or the oil is cold is paramount, and this phase of burner design and operation has been given the closest attention, as faulty ignition is more to be feared than improper operation once the flame is established.

The effect of the air and oil setting is important, since it may be necessary in some instances to adjust for greater excess air than is otherwise required in order to get a mixture suitable for certainty of ignition.

Recent research[1] at Yale University, conducted in coöperation with the A.S.H.V.E. Research Laboratory and the *American Oil Burner Association*, reveals that the various methods of operation (*i.e.*, on and off, high and low, and graduated) all have potential advantages and disadvantages and that a choice in any case requires a consideration of the heat-absorbing characteristics of the boiler in which the burner is to operate. From the standpoint of efficiency of operation it seems that there is little choice if at the maximum setting of all burners the boiler efficiency were at its maximum value. If, on the other hand, the boiler were operating beyond its point of maximum efficiency, then it appears that the graduated type

[1]Intermittent Operation of Oil Burners, by L. E. Seeley and J. H. Powers (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

might show better results. The following additional factors should be considered:

1. The intermittent type by having only one setting might be set for efficient combustion at that point.

2. An intermittent burner should be set for a higher total heat output than either of the other two types in order to get the acceleration necessary when heat is required. This may in some cases give less economical performance due to the increased boiler load.

3. An interruption in electric current might in some instances be troublesome with continuously operating burners where manual ignition is employed.

4. The continuously operating burners must have a minimum fuel setting low enough to prevent overheating in mild weather or during the summer if the boiler is used for domestic hot water.

5. Electrical operating costs must be considered, but must be based upon known power requirements. The power requirements of some burners will be several times as high as others so any generalization on operating costs is futile.

6. Evenness of heat supply will have some influence on uniformity of temperature.

7. Number and cost of controls which reflect in the manufacturing costs.

This entire subject, therefore, is likely to be somewhat perplexing because of the necessity of knowing, and the difficulty in determining, the efficiency characteristics of many heating boilers. Selections of oil burners on the basis of their manner of operation will probably be largely a matter of preference. The advent of special boilers for oil burning will provide the engineer with the opportunity for greater discrimination.

Temperature Control, Protective Devices

Domestic oil burners are controlled directly from the change in temperature of a designated control room (usually the living room, dining room or hall), and by temperature or pressure variations in the boiler. Oil-burner installations put in only a few years ago were simplified to the extent of having a single control element—the room thermostat—that started and stopped the burner. The modern installation provides, in addition, electrical devices inter-wired with the control system to insure against poor operation and to guard against troubles brought on by the characteristics of the heating plant.

One control system provides an instrument actuated by two temperature bulbs, one placed in the outdoor air and the other in a designated part of the heating system. The control is actuated by both bulbs and is designed to maintain the heating medium at a temperature to suit the variations in outdoor temperature, the lower the outdoor temperature the higher the temperature of the heating medium. Other devices have been developed to maintain a certain minimum temperature that will effectively prevent the downward window currents of cold air from reaching and traveling across the floor, regardless of the room thermostat.

Owing to the comparative intensity of heat production with a burner, a boiler with limited water storage above the crown sheet might pass steam to the radiator system so rapidly, at starting, that the sheet would be uncovered, with probable damage to the boiler structure. A low-water safety can be so wired into the system that the burner will be stopped before the water level is reduced to the danger point, or a boiler feed can

be installed to add water to the boiler to maintain a safe level, instead of stopping the burner. Either or both should form part of a first-class installation.

Again, with either steam or water systems, the burner control can be inter-wired with a thermostatic device having its temperature element introduced into the boiler near the top, its function being to limit the maximum temperature of water or pressure of steam so the burner will be shut off before dangerous temperatures or pressures are reached. Windows of the room in which the thermostat is located are sometimes opened to *air out* the house in the morning, and if they are not closed promptly, the burner will operate continuously and possibly develop temperature and pressure conditions that might be detrimental to the boiler. This is where the safety device can be used to offset the carelessness of the human being.

Safety controls have been developed for intermittent burners to guard against failure of ignition and in some instances against momentary flame failures. In general regulatory devices are well developed and dependable. Otherwise the domestic oil burner probably would not have been possible.

For further information on temperature control with oil burners, see Chapter 14.

Boilers for Domestic Oil Burners[2]

Boilers used with domestic installations may be those designed for solid fuel or those designed for liquid fuel. The latter are coming to the fore with great rapidity as they usually have greatly increased secondary surface. Many are of copper or steel tube design. Increased efficiencies of 5 per cent to 15 per cent are often obtainable with boilers designed especially for liquid fuel.

It is possible to go to extremes in providing secondary surfaces sufficient to reduce flue temperatures to the order of 250 F to 300 F, with the result that the added resistance through the flues may necessitate the use of a booster fan to insure sufficient draft. It is difficult to obtain satisfactory efficiencies with boilers having little or no secondary surfaces, where the hot products of combustion pass almost immediately from the combustion chamber to the flue; in fact a high efficiency is unlikely with any fuel under such conditions, and the intermittent burner is especially at a disadvantage because of its characteristic development of heat at a high rate while it is operating.

It is essential that the flame produced by an oil burner, especially where it is strongly luminous, be kept from contact with the water-backed surfaces of the combustion chamber, and to this end bricking or its equivalent must be provided in most cases. Where a burner fires through the ash pit, doorframe bricking must protect the unbacked surfaces of the ash pit. The same fire bricking constitutes the actual combustion chamber for the burner flame, and materially increases the combustion volume for a given boiler.

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[2]For additional information on this subject, refer to Study of Performance Characteristics of Oil Burners and Low-Pressure Heating Boilers, by L. E. Seeley and E. J. Tavanlar (A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Installation

The intelligence and care with which a burner is installed largely determines the satisfaction that will result from its operation. Two plans for the installation of burners are in general use. In the first, the dealer makes all installations. In the other, sales agencies function only to make sales, and the installation for as many as twenty such sales offices is done by a centrally located installation force, usually factory controlled.

Some burners are adjusted for oil rate by means of a blind needle valve that can only be operated with a special wrench; others, by changing the size of the orifice; others, by a combination of orifice size and pressure. In any event, changes in the firing rate, involving careful air and draft adjustment to match the oil rate, should be made by only a trained man,

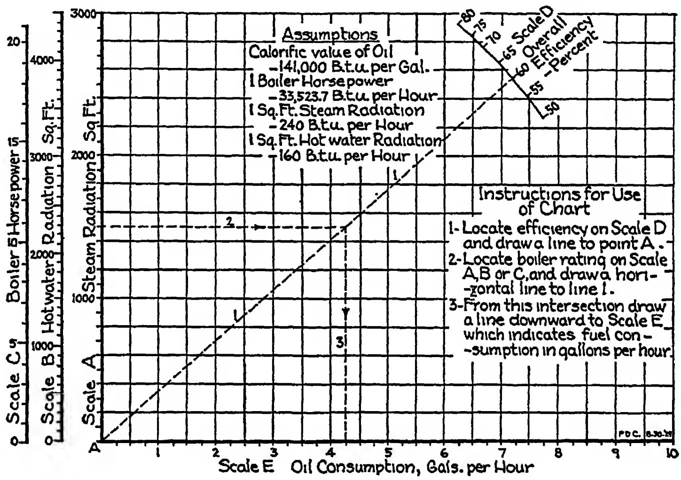

FIG. 1. FULL LOAD RATE OF OIL CONSUMPTION FOR HEATING BOILERS

preferably with the aid of an Orsat test set so that the degree of combustion efficiency can be determined. It is practically impossible to set a burner flame by eye, although that has been general practice in the past. The industry is turning to the Orsat and, as a result, more domestic burners are operating at from 9 per cent to 12 per cent CO_2, representing a higher efficiency combustion than at 5 per cent to 8 per cent, as frequently is the case where the burner is adjusted by eye.

Air for Combustion

It is essential that the basement, or at least that portion used as a boiler room, be open to the outside air, in order that sufficient air be available for combustion. Frequently a case of poor operation will be found where a test with a draft gage made by inserting the tube through the keyhole of the outer door will show that there is a partial vacuum in the basement when the burner is running, all of the combustion air coming through the keyhole and minute cracks. A simple remedy is to cut an inch from the bottom of the outer door.

In order to achieve satisfactory heating at the lowest cost, careful consideration should be given to oil, air and draft adjustment. The oil adjustment should be determined from the total heat requirements to be met. The heat loss of the building plus an allowance for piping plus 20 to 25 per cent for pick-up establishes the maximum output required from the boiler. Fig. 1 indicates the oil required in gallons. Piping allowances will usually vary between 25 and 10 per cent, decreasing with an increase in the size of the building.

With the oil rate thus fixed, the air and draft should be set to give efficient combustion (that is, 10 to 12 per cent CO_2). The furnace draft should be set reasonably low and should be maintained constant by means of an automatic draft regulator. Without this the air supply will fluctuate, causing uneven performance. A check should be made to insure that ignition will be satisfactory under all conditions. An oil burner of the continuous type might dispense with all or part of the pick-up allowance due to the nature of its operation. Careful adjustment will provide ample heat output under all conditions, will minimize the load on the boiler, and will establish the most favorable conditions for intermittent operation.

An essential element in the satisfactory operation of domestic oil burners is the provision for maintenance and service for the burners. What might be called *emergency service* for mechanical or electrical failure of the burner has rapidly diminished during the last few years until a level has been reached where groups of 100 to 1000 burners in a community consistently will require an average of not more than one call per burner per heating season. Maintenance service is coming into general practice where, for a fixed annual payment, regular inspection is made of the burner, and faulty operation corrected before the burner becomes inoperative. This service may contemplate entire overhauling of the burner during the summer, and may include annual cleaning of the boiler flues with a specially designed vacuum cleaner.

Domestic Hot Water Supply

Provision may be made for heating domestic water through exchange heaters attached to the boiler, in which water is maintained at a fixed temperature or steam at a set pressure during the entire year. The flow of water or steam to the radiators is controlled by electrically-operated valves, which remain closed during warm weather and open (through the functioning of the room thermostat) when heat is required in the house. The room thermostat either causes heat to be produced by starting the burner when the room temperature drops to a predetermined point, or closes the circuit of the motor by operating a valve in the flow line of the heating system, the motor opening the water or steam valve and permitting water or steam immediately to flow to the radiators. When the flow in a water heating system is sluggish, the room thermostat also can start the motor of a circulation pump, thereby decreasing the time required to bring the room temperature up to the desired point.

It is usual in small steam heating systems to dispense with the motor-operated valve and by means of an aquastat maintain the boiler water at a constant temperature but well below the steaming temperature (*i.e.*,

140 to 180 F). The lowest temperature setting that will produce sufficiently hot water will be the most economical. The aquastat will always function in such a way as to maintain this temperature except when the room thermostat calls for heat, which means that a call for steam can be more quickly obtained.

Another type of control valve available for hot water systems is thermostatically operated so as to prevent a flow of water to the heating system until the call of the room thermostat for heat raises the water temperature above that normally required for domestic hot water. It should be noted that, except in the case of the graduated burner, the water temperature in the heating system will nearly always reach its maximum, thereby depriving this system to some degree of its natural advantage of modulation.

COMMERCIAL OIL BURNERS

Liquid fuels are used for heating apartment buildings, hotels, public and office buildings, schools, churches, hospitals, department stores, as well as industrial plants of all kinds. Contrary to domestic heating, convenience seldom is a dominating factor, the actual net cost of heat production usually controlling the selection of fuel. Some of the largest office buildings have been using oil for many years. Many department stores have found that floor space in basements and sub-basements can be used to better advantage for merchandising wares, and credit the heat producing department with this saving.

Wherever possible, the boiler plant should be so arranged that either oil or solid fuel can be used at will, permitting the management to take advantage of changes in fuel costs if any occur. Each case should be considered solely in the light of local conditions and prices.

Burners for commercial heating may be either large models of types used in domestic heating, or special types developed to meet the conditions imposed by the boilers involved. Generally speaking, such burners are of the mechanical or pressure atomizing types, the former using rotating cups producing a horizontal torch-like flame. As much as 350 gal of oil per hour can be burned in these units, and frequently they are arranged in multiple on the boiler face, from two to five burners to each boiler.

The larger installations are nearly always started with a hand torch, and are manually controlled, but the use of automatic control is increasing, and completely automatic burners are now available to burn the two heaviest grades of oil. Nearly all of the smaller installations, in schools, churches, apartment houses and the like, are fully automatic.

Because of the viscosity of the heavier oils, it is customary to heat them before transferring by truck tank. It also has been common practice to preheat the oil between the storage tank and the burner, as an aid to movement of the oil as well as to atomization. This heating is accomplished by heat-transfer coils, using water or steam from the heating boiler, and heating the oil to within 30 deg of its flash point.

Unlike the domestic burner, units for large commercial applications frequently consist of atomizing nozzles or cups mounted on the boiler front with the necessary air regulators, the pumps for handling the oil

and the blowers for air supply being mounted in sets adjacent to the boilers. In such cases, one pump set can serve several burner units, and common prudence dictates the installation of spare or reserve pump sets. Pre-heaters and other essential auxiliary equipment also should be installed in duplicate.

Boiler Settings

As the volume of space available for combustion is the determining factor in oil consumption, it is general practice to remove grates and extend the combustion chamber downward to include or even exceed the ash-pit volume; in new installations the boiler should be raised to make added volume available. Approximately 1 cu ft of combustion volume should be provided for every developed boiler horsepower, and in this volume from 1.5 to 2 lb of oil can properly be combusted. This corresponds to a maximum liberation of about 38,000 Btu per cubic foot per hour. There are indications that at times much higher fuel rates may be satisfactory. This in turn suggests that the value of 38,000 Btu per cubic foot per hour should be used with reasonable judgment. For best results, care should be taken to keep the gas velocity below 40 ft per second. Where checkerwork of brick is used to provide secondary air, good practice calls for about 1 sq in. of opening for each pound of oil fired per hour. Such checkerwork is best adapted to flat flames, or to conical flames that can be spread over the floor of the combustion chamber. The proper bricking of a large or even medium sized boiler for oil firing is important and frequently it is advisable to consult an authority on this subject. The essential in combustion chamber design is to provide against flame impingement upon either metallic or fire-brick surfaces. Manufacturers of oil burners usually have available detailed plans for adapting their burners to various types of boilers, and such information should be utilized.

GAS-FIRED APPLIANCES

The increased use of gas for house heating purposes has resulted in the production of such a large number of different types of gas-heating systems and appliances that today there is probably a greater variety of them than there is for any other kind of fuel.

Gas-fired heating systems may be classified as follows:

- I. Gas-Designed Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water, and vapor boilers.
    2. Warm air furnaces.
  - B. Unit Heating Systems.
    1. Warm air floor furnaces.
    2. Industrial unit heaters.
    3. Space heaters.
    4. Garage heaters.
- II. Conversion Heating Systems.
  - A. Central Heating Plants.
    1. Steam, hot water and vapor boilers.
    2. Warm air basement furnaces.

The majority of these systems are supplied with either automatic or manual control. Central heating plants, for example, whether gas designed or conversion systems, may be equipped with room temperature control, push button control, or manual control.

Although no exact rules can be prescribed as to the field best covered by each of the foregoing systems, each installation will have problems pointing more or less directly to some particular type of heating equipment.

Gas-Fired Boilers

Information on gas-fired boilers will be found in Chapter 25.

Either snap action or throttling control is available for gas boiler operation. This is especially advantageous in straight steam systems because steam pressures can be maintained at desired points, while at the same time complete cut-off of gas is possible when the thermostat calls for it.

Warm Air Furnaces

There are two general classes of gas-fired warm air furnaces, the gravity furnace which depends upon the natural tendency of heated air to rise, providing the proper circulation of heated air into the room, and the mechanical circulation furnace by which the air to be heated is forced through or drawn through the furnace by means of a fan.

Warm air furnaces are variously constructed of cast iron, sheet metal and combinations of the two materials. If sheet metal is used, it must be of such a character that it will have the maximum resistance to the corrosive effect of the products of combustion. With some varieties of manufactured gases, this effect is quite pronounced. Warm air furnaces are obtainable in sizes from those sufficient to heat the largest residence down to sizes applicable to a single room. The practice of installing a number of separate furnaces to heat individual rooms is peculiar to mild climates, such as that of southern California. Small furnaces, frequently controlled by electrical valves actuated by push-buttons in the room above, are often installed to heat rooms where heat may be desired for an hour or so each day. These furnaces are used also for heating groups of rooms in larger residences. In a system of this type each furnace should supply a group of rooms in which the heating requirements for each room in the group are similar as far as the period of heating and temperature to be maintained are concerned. Bedrooms, living rooms, and dining rooms, often present excellent possibilities for this type of furnace.

The same fundamental principle of design that is followed in the construction of boilers, that is, breaking the hot gas up into fine streams so that all particles are brought as close as possible to the heating surface, is equally applicable to the design of warm air furnaces. The desirability of using an appliance designed for gas, when gas is to be the fuel, applies even more strongly to furnaces than to boilers.

Codes for proportioning warm air heating plants, such as that formulated by the *National Warm Air Heating Association* (see note p. 329), are equally applicable to gas furnaces and coal furnaces. Recirculation should always be practiced with gas-fired warm air furnaces. It not only aids in heating, but is essential to economy. Where fans are used in connection with warm air furnaces for residence heating, it is well to

have the control of the fan and of the gas so coördinated that there will be sufficient delay between the turning on of the gas and the starting of the fan to prevent blasts of cold air being blown into the heated rooms. An additional thermostat in the air duct easily may be arranged to accomplish this.

Floor Furnaces

Warm air floor furnaces are well adapted for heating first floors, or where heat is required in only one or two rooms. A number may be used to provide heat for the entire building where all rooms are on the ground floor, thus giving the heating system flexibility as any number of rooms may be heated without heating the others. With the usual type the register is installed in the floor, the heating element and gas piping being suspended below. Air is taken downward between the two sheets of the double casing and discharged upward over the heating surfaces and into the room. The appliance is controlled from the room to be heated by means of a control lever located near the edge of the register. The handle of the control is removable as a precaution against accidental turning on or off of the gas to the furnace.

Space heaters are generally used for auxiliary heating, but may be, and are in many cases, installed for furnishing heat to entire buildings. Space heaters are quite extensively used for house heating in milder climates such as exist in the South and Southwest. With the exception of wall heaters, they are portable, and can be easily removed and stored during the summer season. Although they should be connected with solid piping it is sometimes desirable to connect them with flexible gas tubing in which case a gas shut-off on the heater is not permitted, and only *A.G.A.* approved tubing should be used.

Space Heaters

Parlor furnaces or *circulators* are usually constructed to resemble a cabinet radio. They heat the room entirely by convection, *i.e.* the cold air of the room is drawn in near the base and passes up inside the jacket around a drum or heating section, and out of the heater at or near the top. These heaters cause a continuous circulation of the air in the room during the time they are in operation. The burner or burners are located in the base at the bottom of an enclosed combustion chamber. The products of combustion pass up around baffles within the heating element or drum, and out the flue at the back near the top. They are well adapted not only for residence room heating but also for stores and offices.

Radiant heaters make admirable auxiliary heating appliances to be used during the occasional cool days at the beginning and end of the heating season when heat is desired in some particular room for an hour or two. The radiant heater gives off a considerable portion of its heat in the form of radiant energy emitted by an incandescent refractory that is heated by a Bunsen flame. They are made in numerous shapes and designs and in sizes ranging from two to fourteen or more radiants. Some have sheet-iron bodies finished in enamel or brass while others have cast-iron or brass frames with heavy fire clay bodies. An atmospheric burner is supported near the center of the base, usually by set screws at each end. Others

have a group of small atmospheric burners supported on a manifold attached to the base. Most radiant heaters are supported on legs and are portable; however, there are also types which are encased in a jacket which fits into the wall with a grilled front, similar to the ordinary wall register. Others are encased in frames which fit into fireplaces.

Gas-fired steam and hot water radiators are popular types of room heating appliances. They provide an economical form of heating apparatus for intermittently heated spaces such as stores, small churches and some types of offices and apartments. They are made in a large variety of shapes and sizes and are similar in appearance to the ordinary steam or hot water radiator connected to a basement boiler. A separate combustion chamber is provided in the base of each radiator and is usually fitted with a one-piece burner. They may be secured in either the vented or unvented types, and with steam pressure, thermostatic or room temperature controls.

Warm air radiators are similar in appearance to the steam or hot water radiators. They are usually constructed of pressed steel or sheet metal hollow sections. The hot products of combustion circulate through the sections and are discharged out a flue or into the room, depending upon whether the radiator is of the vented or unvented type.

Garage heaters are usually similar in construction to the cabinet circulator space heaters, except that safety screens are provided over all openings into the combustion chamber to prevent any possibility of explosion from gasoline fumes or other gases which might be ignited by an open flame. They are usually provided with automatic room temperature controls and are well suited for heating either residence or commercial garages.

Conversion Burners

Residence heating with gas through the use of conversion burners installed in coal-designed boilers and furnaces represents a common type of gas-fired house heating system, especially in natural gas territories. In many conversion burners radiants or refractories are employed to convert some of the energy in the gas to radiant heat. Others are of the blast type with luminous flames, operating without refractories. In each case an attempt is made to transfer the majority of the heat from the gas to the medium to be heated within the fire pot itself because of the low heat transfer that takes place in the flue passages.

Many conversion units are equipped with sheet metal secondary air ducts which are inserted through the ash-pit door. The duct is equipped with automatic air controls which open when the burners are operating and close when the gas supply is turned off. This prevents a large part of the circulation of cold air through the combustion space of the appliance when not in operation. By means of this duct the air necessary for proper combustion is supplied directly to the burner, therefore making it possible to reduce the amount of excess air passing through the combustion chamber.

Conversion units are made in many sizes both round and rectangular to fit different types and makes of boilers and furnaces. They may be secured with manual, push button, or room temperature control.

Sizing Gas-Fired Heating Plants

While gas-burning equipment can be and usually is so installed as to be completely automatic, maintaining the temperature of rooms at a predetermined and set figure, there are in use installations which are manually controlled. Experience has shown that in order to effectively overcome the starting load and losses in piping, a manually-controlled gas boiler should have an output as much as 100 per cent greater than the equivalent standard cast-iron column radiation which it is expected to serve.

Boilers under thermostatic control, however, are not subject to such severe pick-up or starting loads. Consequently, it is possible to use much lower selection or safety factors. A gas-fired boiler under thermostatic control is so sensitive to variations in room temperatures that in most cases a factor of 25 per cent is sufficient for pick-up load.

Table 1. Selection Factors for Gas Boilers

Equivalent Cast Iron Steam Radiation (Square Feet of 240 Btu Each)	Selection Factor (Per Cent)
500	56.0
800	54.0
1,200	51.0
1,600	48.0
2,000	45.0
3,000	42.5
4,000 and over	40.0

The factor to be allowed for loss of heat from piping, however, must vary somewhat, the proportionate amount of piping installed being considerably greater for small installations than for large ones. Consequently, a selection factor for thermostatically controlled boilers must be variable. Table 1 gives selection factors to be added to the installed steam radiation under thermostatic control. They have been established by experience and are recommended by the *American Gas Association.*

The same factors may be used in determining the gas demand for which conversion burners installed in steam or hot water boilers should be set. Multiplying the equivalent direct heating surface (radiation) by 240 and adding the appropriate percentage from Table 1, and then dividing by the heat value of the gas and by the heating efficiency (see discussion of heating efficiencies in Chapter 29), gives the proper hourly rate of gas consumption. However, inadequate boiler heating surface for gas burning, often encountered in coal-designed boilers converted to gas, may necessitate operation at a lesser demand, resulting in much slower pick-up and less margin of safety for piping loss.

Appliances used for heating with gas should bear the approval seal of the *American Gas Association* Testing Laboratory. Installations should be made in accordance with the recommendations shown in the publications of that association.

Ratings for Gas Appliances

Since a gas appliance has a heat-generating capacity that can be pre-

dicted accurately to within 1 or 2 per cent, and since this capacity is not affected by such things as condition of fuel bed and soot accumulation, makers of these appliances have an opportunity to rate their product in exact terms. Consequently all makers give their product an hourly Btu output rating. This is the amount of heat that is available at the outlet of a boiler in the form of steam or hot water, or at the bonnet of the furnace in the form of warm air. The output rating is in turn based upon the Btu input rating which has been approved by the *American Gas Association* Testing Laboratory and upon an average efficiency which has been assigned by that association.

In the case of boilers, the rating can be put in terms of square feet of equivalent direct radiation by dividing it by 240 for steam, and 150[3] for water. This gives what is called the *American Gas Association* rating, and is the manner in which all appliances approved by the *American Gas Association* Laboratory are rated. To use these ratings it is only necessary to increase the calculated heat loss or the equivalent direct radiation load by an appropriate amount for starting and piping, and to select the boiler or furnace with the proper rating.

The rating given by the *American Gas Association* Laboratory is not only a conservative rating when considered from the standpoint of capacity and efficiency, but is also a safe rating when considered from the standpoint of physical safety to the owner or caretaker. The rating that is placed upon an appliance is limited by the amount of gas that can be burned without the production of harmful amounts of carbon monoxide. This same limitation applies to all classes of gas-consuming heating appliances that are tested and approved by the Laboratory. Gas boilers are available with ratings up to 14,000 sq ft of steam, while furnaces with ratings up to about 500,000 Btu per hour are available. (See Chapter 23).

Installation Features

One feature of the piping installation that adds to the satisfactory service rendered by gas boilers is provision for adequate and rapid venting of the air from steam heating systems. If air leaks into the steam distribution system during the period that the gas is turned off, and then vents out slowly when the thermostat calls for heat, the result will be a further cooling of the premises between the time that the thermostat calls for heat and the time that steam reaches the radiators. A freely venting steam or vapor system gives maximum economy and minimum temperature variation. When gas boilers are attached to existing heating plants, it is good practice to check the effectiveness of the venting devices and if necessary to replace them with more effective ones that will prevent the return of air into the heating system, and also to check the tightness of the piping.

Frequently when a coal boiler is already installed in a home, it is expedient to leave the coal boiler in place, and to cross-connect the gas boiler with it. Where gas heating is new to the community, it pro-

[3]A value of 160 for the heat emission of hot water radiators is used by many engineers. The actual heat emission, however, depends on the temperature of the water and of the surrounding air. See Chapters 30 and 33.

duces a more secure feeling in the customer's mind when putting in gas-fired house-heating equipment, if he knows that he can burn coal at any time he may desire. For steam or vapor installations, it is desirable to have the water line in both boilers at the same level.

Chapter 29

FUEL CONSUMPTION

Heat Loss, Calorific Values, Heating Efficiencies, Non-Heating Periods, Heat Capacity of Buildings, Miscellaneous Factors, Degree-Day Method, Rough Approximations, Relative Heating Costs

TO predict the amount of fuel likely to be consumed in heating a building during a normal heating season, it is necessary to know the total heat requirements of the building and the utilization factor of the fuel. The accuracy of the estimate will depend on the ability to select these values and on the care taken in making allowances for other variable factors.

Fuel requirements[1] are given by the following general equation:

$$F = \frac{H \times (t - t_a) \times N}{(t - t_o) \times C \times E} \tag{1}$$

where

F = quantity of fuel required for a heating season.
N = number of hours of heating season corresponding to average temperature, t_a.
t = inside temperature, degrees Fahrenheit.
t_a = average outside temperature, degrees Fahrenheit.
t_o = outside design temperature, degrees Fahrenheit.
H = calculated heat loss of building based on outside temperature of t_o, Btu per hour.
C = calorific value of one unit of fuel, the unit being the same as that on which F is based.
E = efficiency of utilization of fuel, per cent.

HEAT LOSS

The hourly heat loss (H) is equal to the sum of the transmission losses (H_t) and the infiltration losses (H_i) of the rooms or spaces to be heated, and the total equivalent heating surface required is equal to $\frac{H}{240 \text{ sq ft}}$. In estimating the fuel consumption of a building of more than one room divided by walls or partitions, it is not correct to use the calculated heat loss of the building without making the proper allowance for the fact that the heating load at any time does not involve the sum of the infiltration losses of all of the heated spaces of the building but only part of the infiltration losses. This is explained in Chapter 6.

It is sufficiently accurate in most cases to consider only half of the total infiltration losses of a building having interior walls and partitions, and the value of H in Equation 1 would, under these conditions, be equal to

[1]For further information on this subject see Estimating Fuel Consumption, by Paul D. Close, (*Heating, Piping and Air Conditioning*, May, 1931).

$H_t + \frac{H_i}{2}$. If a building has no interior walls or partitions, whatever air enters through the cracks on the windward side must leave through the cracks on the leeward side, and only half of the total crack should be used in computing the infiltration for each side and end of the building. Under these conditions it is sufficiently accurate to use the total calculated heat loss (H) for the building. If the average wind velocity during the heating season differs from that upon which H_i was derived, the value of H should be corrected accordingly.

Of course, where the required heating surface is estimated by empirical or rule-of-thumb methods, refinements in approximating fuel consumption are not warranted, but rule-of-thumb methods often lead to unsatisfactory results and should be avoided in heating work where more accurate methods are available. It should be emphasized that the value of H in Equation 1 is the total heat loss of the building after making the proper allowance for infiltration.

CALORIFIC VALUES AND HEATING EFFICIENCIES

The calorific values of fuel oils and gas can be ascertained with reasonable accuracy. The values for various grades of oil are given in Table 2, Chapter 27. The calorific value of gas can always be obtained from the local utility company. Values for natural gas are given in Table 3, Chapter 27; manufactured gas usually has a calorific value of about 535. Coals have a larger range and may vary for the same type of coal, depending on its ash content. For general purposes where specific data are lacking, values can be taken from the top curve of Fig. 1, Chapter 27.

To decide on the correct efficiency to use is a more difficult matter, particularly if the estimate is being made without a full knowledge of the equipment for burning the fuel and the care the furnace will receive. Efficiencies usually are given in the catalogs of manufacturers of furnaces and boilers, but these values are obtained under test conditions and do not allow for poor attendance, defects in installation, or poor draft. On the other hand, such efficiencies assume that all the heat radiated from the outside of the heaters or casings as sensible heat of the flue gases is lost, whereas, if the heater is installed in the building being heated, a considerable portion of these losses may help to heat the building[2]; how much of this it is legitimate to use in increasing the value of E will depend on whether H included the heat losses in the cellar, and on the construction of the chimney. Except for an interior chimney, the heat transferred through the chimney wall to the building will be very small. Chimney allowances should be greater for lower test efficiencies. Thus an insulated furnace will give a high efficiency on test but will not heat the cellar. A modern gas furnace will have a high efficiency with a correspondingly low flue gas temperature and hence there will be very little heat from the flue pipe.

For great exactitude the value for E should take care of inefficiency in the heat distribution in the building because of such losses as excessive heating of the walls behind the radiators and excessive stratification. It

[2]Analysis of the Over-All Efficiency of a Residence Heated by Warm Air, by A. P. Kratz and J. F. Quereau (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

TABLE 1. DEGREE-DAYS FOR CITIES IN THE UNITED STATES AND CANADA[a]

Col. A State	*Col. B* City	*Col C* Degree-Days	*Col. A* State	*Col B* City	*Col. C* Degree-Days
Ala.	Birmingham	2,408	Nev.	Reno	5,891
	Mobile	1,471	N. H.	Concord	6,852
Ariz.	Flagstaff	7,145	N. J.	Atlantic City	5,175
	Tucson	1,845		Trenton	4,934
Ark.	Hot Springs	2,665	N. M.	Santa Fe	6,063
	Little Rock	2,811	N. Y.	Albany	6,889
Calif.	Los Angeles	1,504		Buffalo	6,821
	San Francisco	3,264		New York	5,348
Colo.	Colorado Springs	6,553	N. C.	Raleigh	3,234
	Denver	5,873		Wilmington	2,302
Conn.	New Haven	5,895	N. D.	Bismarck	8,498
D. C.	Washington	4,626	Ohio	Cincinnati	4,702
Fla.	Jacksonville	890		Cleveland	6,154
Ga.	Atlanta	2,891		Columbus	5,323
	Savannah	1,490	Okla.	Oklahoma City	3,613
Idaho	Boise	4,558	Ore.	Portland	4,468
	Lewiston	4,924		Salem	4,629
Ill.	Chicago	6,315	Pa.	Philadelphia	4,855
	Springfield	5,370		Pittsburgh	5,235
Ind.	Evansville	4,164	R. I.	Providence	6,014
	Indianapolis	5,297	S. C.	Charleston	1,769
Iowa	Des Moines	6,373		Spartanburg	3,257
	Sioux City	7,023	S. D.	Sioux Falls	7,683
Kan.	Dodge City	5,034	Tenn.	Memphis	2,950
	Topeka	5,301		Nashville	3,578
Ky.	Lexington	4,616	Texas	Austin	1,578
	Louisville	4,180		Dallas	2,455
La.	New Orleans	1,023		Houston	1,157
Me.	Eastport	8,531		San Antonio	1,202
	Portland	7,012	Utah	Logan	6,735
Md.	Baltimore	4,333		Salt Lake City	5,553
Mass.	Springfield	6,464	Vt.	Burlington	7,620
	Boston	6,145	Va.	Fredericksburg	4,243
Mich.	Detroit	6,494		Norfolk	3,349
	Marquette	8,692		Richmond	3,725
Minn.	Duluth	9,480	Wash.	Seattle	4,868
	Minneapolis	7,851		Spokane	6,353
Miss.	Vicksburg	1,822	W. Va.	Morgantown	5,016
Mo.	Kansas City	5,202		Parkersburg	4,884
	St. Louis	4,585	Wis.	Fond du Lac	7,612
Mont.	Billings	7,115		Green Bay	7,823
	Havre	8,699		La Crosse	6,690
Neb.	Lincoln	6,231		Milwaukee	7,372
	Omaha	6,128	Wyo.	Cheyenne	7,462

Province	City	Degree-Days	Province	City	Degree-Days
B. C.	Victoria	5,777	Ont.	Toronto	7,732
	Vancouver	5,976	Que.	Montreal	8,705
	Kamloops	6,724		Quebec	8,628
Alb.	Medicine Hat	8,152	N. B.	Fredericton	9,099
Sask.	Qu'Appelle	11,261	N. S.	Yarmouth	7,694
Man.	Winnipeg	11,166	P. E. I.	Charlottetown	8,485
Ont.	Port Arthur	10,803			

[a]From *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association*. These degree-days are based on daily mean temperatures. Base, 65 F.

is preferable, however, to include these losses in the value of H, and to limit E to the fuel burning equipment.

Automatic fuel burning equipment, whether for coal, oil or gas, will tend to save fuel and will therefore produce a higher efficiency if thermostatically controlled, but on the other hand automatic equipment tends to make the householder prolong his heating season and to maintain a higher temperature in the house in the early fall and late spring.

NON-HEATING PERIODS

Obviously, the theoretical fuel consumption will be reduced considerably by not operating the heating plant at night. Allowance for this may be made in either of two ways: (1) by estimating the average inside temperature (t), or (2) by arbitrarily assuming a certain reduction in the fuel consumption.

The first procedure is, of course, the more accurate. If, for example, the daytime temperature is to be 70 F, and the temperature from 12 midnight to 6 a.m. is to be maintained by thermostatic control at 50 F, then the average daily inside temperature (t) will be $\frac{18 \times 70 + 6 \times 50}{24}$ or 65 F. Strictly speaking, this average inside temperature would only apply when the outside night temperature averages below 50 F, but this fact usually is not of sufficient importance to warrant consideration. If the average outside temperature during the heating season is 30 F, the fuel saving would be approximately $100 \times \frac{70 - 65}{70 - 30}$ or 12.5 per cent. In this case, the additional saving in fuel due to the cooling of the air and structural materials to 50 F would be offset by the heating-up load in the morning.

As to the second procedure, it may be arbitrarily assumed that a saving in the fuel consumption of from 10 to 30 per cent (depending on conditions) will result if the heat is shut off after working hours, and the building heated to the required temperature during the period of occupancy each day. This, of course, is a general statement and wherever possible the average temperature should be estimated from the proportionate lengths of the occupancy and non-occupancy periods and the corresponding temperatures for these periods. Any deviation from the assumed inside temperature will result in a variation in the estimated fuel consumption.

HEAT CAPACITY OF BUILDINGS

The heat required to warm the cold building and contents is a factor to be considered. Under certain conditions, the cooling of the structure and contents will, to some extent, compensate for the heat required to rewarm the building. For example, if the building is under thermostatic control and the day and night temperatures are say 70 F and 50 F, respectively, there will be a period during which no heat will be called for while the building is cooling to 50 F, and the saving resulting therefrom will correspond to the additional heat required to bring the building and contents back to the daytime temperature. If in estimating the fuel consumption the average daily inside temperature is based on the proper day and night temperatures and periods, the heat required to warm the structure may be neglected.

Where irregular conditions are involved it may be desirable to actually calculate the fuel required to warm the building structure and contents for the number of times during the heating season the heating plant would not be in operation and to add this quantity to the fuel required for the number of hours during which the building is heated. The greater the heat capacity of the structure the greater will be the relative importance of this item. For structures of low heat capacity, such as frame buildings, this factor usually may be neglected.

Example 1. A small factory building located in Philadelphia is to be heated to 60 F between the hours of 7 a.m. and 7 p.m., and to 50 F during the remaining hours. The calculated hourly heat loss, based on a design temperature of − 6 F, is 500,000 Btu. If coal having a calorific value of 12,500 Btu per pound is fired, and the over-all heating efficiency is assumed to be 60 per cent, how many tons of coal will be required for a normal heating season, neglecting other heat sources and any loss of heat through open windows?

Solution. Since there are no partitions in this building, the entire heat loss is considered. The average outside temperature during the heating season (t_a) is 41.9 F (see Table 2, Chapter 7); t = 60 F; N = 5040; H = 500,000; $(t - t_o)$ = 66 F; C = 12,500; E = 0.60. Substituting these values in Equation 1 and dividing by 2000 to change to tons:

$$F = \frac{500{,}000 \times 18.1 \times 5040}{66 \times 12{,}500 \times 0.60 \times 2000} = 46 \text{ tons of coal}$$

Inasmuch as the building will be heated to 50 F at night, the average inside temperature (at the breathing line) will be 55 F, and the percentage saving will be $\frac{60 - 55}{60 - 41.9}$ = 0.276 or 27.6 per cent. The net fuel consumption will therefore be 46 − 0.276 × 46 or 33.3 tons.

MISCELLANEOUS FACTORS

There are many factors which would be likely to affect the theoretical fuel requirements of a building, such as the opening of windows, abnormal inside temperatures, other heat sources, sun effect, wind, and rain. In many cases it is difficult to evaluate these factors accurately, particularly in the case of open windows, and the results are correspondingly less accurate. The degree of refinement of the calculations should, of course, be consistent with the conditions involved. If the heat loss from the boiler and piping does not warm the building or is not included in H, the proper allowance should be made. In selecting a boiler, this allowance is frequently assumed to be 25 per cent of the total heat loss of the building, but in estimating fuel requirements, the more accurate procedure of computing the pipe and boiler losses should be used, unless this item is likely to be outweighed by other less tangible factors.

Where temperature control is installed the fuel consumption can obviously be predetermined with greater accuracy than where no such control has been provided. In fact the calculated requirements agree to a remarkable extent in many cases with the actual fuel consumption. This has been particularly true of gas-fired installations, with which effective temperature regulation usually is possible.

OTHER HEAT SOURCES

Where other heat sources are available it is quite often possible to make accurate allowance for the reduction in the fuel consumption resulting

therefrom. These sources include the heat supplied by persons, lights, motors and machinery, and should also be ascertained in the case of theaters, assembly halls and industrial plants. (See Chapter 7). In many cases these heat sources should not be allowed to affect the size of the installation of heating equipment, although they may have a marked effect upon the fuel consumption. In residences this factor usually may be neglected.

DEGREE-DAY METHOD

A very useful unit for estimating fuel consumption, particularly for residences, is the degree-day. (See definition in Chapter 42). Degree-days for various cities in the United States and Canada are given in Table 1. The term degree-day originated in the gas industry and was later standardized by the *American Gas Association*[3].

The base of 65 F is used for an inside temperature of 70 F. This base was chosen because it was demonstrated, by means of data collected from numerous installations, that heat is seldom supplied to a residence when the outdoor temperature is greater than 65 F[4]. It was also found that the fuel consumed varied almost directly with the difference between 65 F and the outside temperature.

If the inside temperature were maintained at 70 F throughout the 24 hours of the day, then the base of 65 F would probably be in error. It must be borne in mind, however, that although the temperature head is the difference between the inside temperature of say 70 F, and the outside temperature, a lower temperature than 70 F will usually be maintained at night and the base of 65 F will therefore allow for this condition. As already indicated, a temperature of 50 F from midnight to 6 a.m. will reduce the 24-hour average from 70 to 65 F. It is important to note that the degree-day applies specifically to an inside temperature of 70 F, which is the usual temperature for residences, and it should also be noted that allowance is automatically made for the lower night-time temperature, although this allowance is constant for any given locality.

In Equation 1, the quantity $(t - t_a) \times N$ is equivalent to the number of degree-days (D) in a heating season multiplied by 24, when the average daily value of t is 65 F. Therefore

$$(t - t_a) \times N = 24\,D \tag{2}$$

Substituting the value of $(t - t_a) \times N$ from Equation 2 in Equation 1, the following general formula for an average daily inside temperature of 65 F, which is approximately equivalent to an inside daytime temperature of 70 F for residences, is obtained:

$$F_d = \frac{24\,HD}{(t - t_o) \times C \times E} \tag{3}$$

Example 2. The calculated hourly heat loss of a residence located in Chicago is 127,000 Btu, which includes 28,000 Btu for infiltration. The design temperatures are − 8 F and 70 F. The normal heating season is assumed to be 210 days (5,040 hours) and the average temperature during this period is 36.4 F (see Table 2, Chapter 7). The

[3]See *Industrial Gas Series, House Heating (third edition)* published by the *American Gas Association.*
[4]See also Iso-degree-day map and charts developed by P. E. Fansler for coal, oil and gas.

building is to be heated with oil fuel having a calorific value of 141,000 Btu per gallon. The heating efficiency is assumed to be 70 per cent. Thermostatic control is to be used and a temperature of 55 F is to be maintained from 11 p.m. to 7 a.m. How many gallons of oil will be required during a normal heating season if the loss of heat through open windows is neglected?

Solution. The maximum hourly heat loss will be $127{,}000 - \frac{28{,}000}{2} = 113{,}000$ Btu $= H$. Substituting the proper values in Equation 1:

$$F = \frac{113{,}000 \times (70 - 36.4) \times 5040}{141{,}000 \times 0.70 \times [70 - (-8)]} = 2486 \text{ gal of oil.}$$

The average inside temperature will be $\frac{70 \times 16 + 55 \times 8}{24} = 65$ F

and the fuel saving due to this fact will be $\frac{70 - 65}{70 - 36.4} = 0.149$ or 14.9 per cent.

Hence, the net fuel consumption will be $2486 - 0.149 \times 2486 = 2116$ gal.

The normal number of degree-days for Chicago is 6315. Substituting in Equation 3 and solving by the degree-day method:

$$F = \frac{113{,}000 \times 6315 \times 24}{78 \times 141{,}000 \times 0.70} = 2225 \text{ gal of oil}$$

No allowance need be made for the average temperature of 65 F since this is taken care of by the selection of a base of 65 F for the degree-day, as already explained. It will be noted that the two methods check within 5 per cent in this case. If the average daily inside temperature in the first solution had been 66.4 F instead of 65 F, the two methods would have checked exactly.

INDUSTRIAL DEGREE-DAY

Since the standard degree-day is intended for an inside temperature of 70 F, it is particularly convenient for solving residence problems. Where the design temperature differs greatly from 70 F, the standard degree-day cannot be accurately applied. Consequently, the industrial degree-day[5] has been developed and values have been derived for two bases, namely 55 F and 45 F, intended for inside temperatures of 60 F and 50 F, respectively.

There is a considerable spread, however, among these three bases, and consequently there would be an appreciable error if the actual basis to be used in a certain case would be approximately midway between any two of the three bases for which degree-day values are at present available. Since the correction cannot be made on a proportionate basis, it would be more accurate in the majority of cases involving inside temperatures other than 70 F, 60 F or 50 F, to apply Equation 1.

APPROXIMATING FUEL REQUIREMENTS

It is sometimes desirable to obtain a rough approximation of the annual fuel consumption. Such approximations may be obtained by using unit factors based on the fuel requirements per square foot (or per 100 sq ft) of *radiation* or per 1000 cu ft of space.

Fig. 1 may be used for rough approximations of coal and oil require-

[5]See *Heating and Ventilating* Degree-Day Handbook.

ments. It should be noted that this figure is given in terms of the fuel consumption per 1000 degree-days per 100 sq ft of equivalent heating surface (steam) based on an emission of 240 Btu per square foot. Unless the *radiation* is calculated with reasonable accuracy, unit factors will be of little value even for rough approximations, since it is obvious that such *radiation* requirements must bear some relationship to the actual heating requirements of the building.

Example 3. Estimate the approximate coal consumption for a building located in New York City in which the calculated heating surface requirements (steam) are 1000 sq ft based on design temperatures of zero and 70 F.

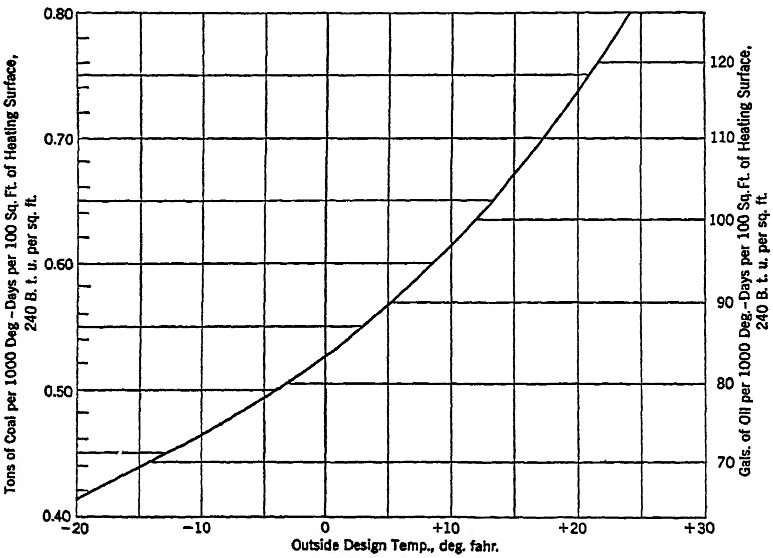

FIG. 1. CURVE FOR OBTAINING ROUGH APPROXIMATION OF ANNUAL FUEL CONSUMPTION IN TONS OF COAL OR GALLONS OF OIL PER 1000 DEGREE-DAYS PER 100 SQ FT OF EQUIVALENT STEAM HEATING SURFACE[a]

[a]This curve is based on heating efficiencies of 60 and 70 per cent for coal and oil, respectively, a calorific value of coal of 13,000 Btu per pound, a calorific value of oil of 14,000 Btu per gallon, an inside temperature of 70 F, and an emission of 240 Btu per equivalent square foot of heating surface (steam), and does not allow for unusual factors which would affect the fuel consumption, such as open windows, week-end shutdowns, etc. For hot water, divide the result obtained by means of this chart by 1.6.

Solution. From Fig. 1, the fuel consumption for a design temperature of zero is 0.53 tons per 1000 degree-days per 100 sq ft of heating surface. Since there are 5348 degree-days in New York City in a normal heating season, the fuel consumption will be approximately $0.53 \times 5.348 \times 10 = 28.34$ tons.

Fig. 2 is taken from the 3rd edition of Industrial Gas Series on House Heating, published by the *American Gas Association,* and indicates the average gas consumption per degree-day for various heat contents. While the fuel consumption in individual cases may vary somewhat from the curve values, these average values are sufficiently accurate for estimating purposes and give very satisfactory results.

The value generally used in the manufactured gas industry for residences is 0.21 cu ft per degree-day per square foot of equivalent steam

radiation (240 Btu) based on the theoretical requirements. A correction for warmer climates is necessary and it is customary to gradually increase the relative fuel consumption below 3,000 degree-days to about 20 per cent more at 1,000 degree-days.

For hot water or warm air heat the fuel consumption is about 0.19 cu ft per degree-day per square foot of equivalent steam *radiation*, that is, per 240 Btu per hour. The actual requirements likewise relatively increase with hot water or warm air systems as the number of degree-days decreases below 3,000. For larger installations, that is, 1,000 sq ft of

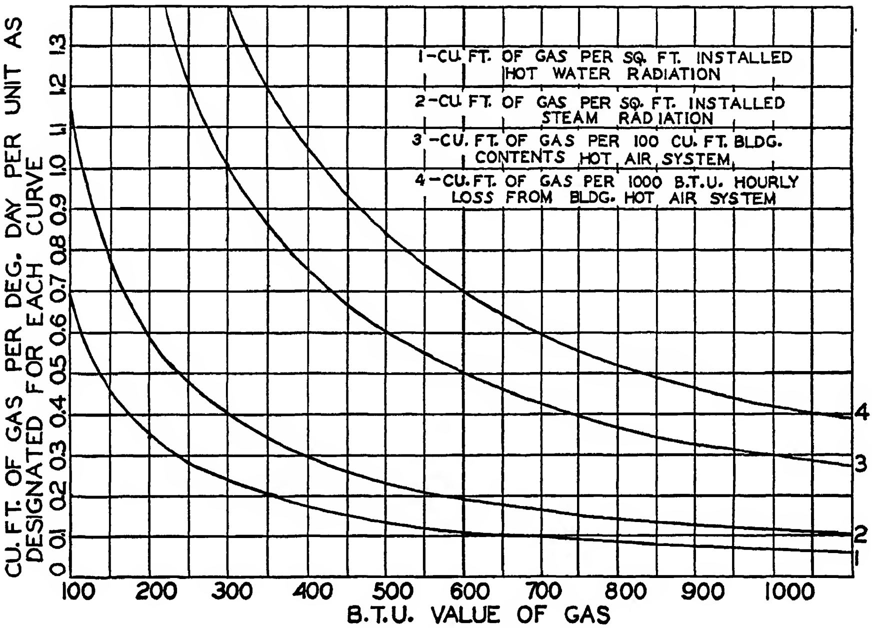

FIG. 2. CHART GIVING GAS REQUIREMENTS PER DEGREE-DAY FOR VARIOUS CALORIFIC VALUES OF GAS AND FOR DIFFERENT HEATING SYSTEMS[a]

[a]This chart is based on an inside temperature of 70 F and an outside temperature of zero. If the *radiation* is installed on the basis of any other temperature difference, multiply the result obtained from this chart by 70, and divide by the actual temperature difference.

theoretical *radiation* and above, there is an increase in efficiency, and a consequent decrease in the fuel consumption per degree-day per square foot of heating surface.

The approximate quantities of steam required in New York City per square foot of heating surface, for various classes of buildings are given in Chapter 36.

The preceding discussion on fuel consumption has dealt with the heating requirements of the building irrespective of any air that may be introduced for ventilation purposes other than the normal infiltration of outside air. The heat required for warming air brought into the building for ventilation may be estimated from data given in Chapters 2 and 22.

RELATIVE HEATING COSTS

A comparison of the relative cost of heating with different fuels can be made with even a fair degree of accuracy only when there is a full knowledge of the equipment which will be used with each fuel, and the efficiency with which each will be operated. When proposing to substitute one fuel for another, the yearly cost with the fuel being used can be obtained. The accuracy of the comparison will depend upon the care taken in estimating the cost of the new fuel with the equipment which will be used.

A convenient basis for comparison of various fuels is the cost per million Btu. The formula used in estimating costs for coal is:

$$X = \frac{500 \times c}{C_c \times E_c} \tag{4}$$

where

X = cost of heating with coal in dollars per million Btu.
c = cost of coal in dollars per ton.
C_c = calorific value of coal, Btu per pound.
E_c = over-all or house efficiency for coal, expressed as a decimal.

Example 4. If coal having a calorific value of 13,000 Btu per pound costs \$10.00 per ton, the cost per million Btu, assuming an efficiency of 60 per cent, will be:

$$X = \frac{500 \times 10}{13{,}000 \times 0.60} = \$0.64$$

The formula used in estimating costs for oil is:

$$Y = \frac{1{,}000{,}000 \times p}{C_o \times W \times E_o} \tag{5}$$

where

Y = cost of heating with oil in dollars per million Btu.
p = cost of oil in dollars per gallon.
C_o = calorific value of oil, Btu per pound.
W = weight of oil per gallon, pounds.
E_o = over-all or house efficiency for oil, expressed as a decimal.

Example 5. If oil having a calorific value of 141,000 Btu per gallon ($C_o \times W$) costs 10¢ per gallon, the cost per million Btu, assuming an efficiency of 70 per cent, will be:

$$Y = \frac{1{,}000{,}000 \times 0.10}{141{,}000 \times 0.70} = \$1.01$$

The formula used in estimating costs for gas is:

$$Z = \frac{1000g}{C_g \times E_g} \tag{6}$$

where

Z = cost of heating with gas in dollars per million Btu.
g = average cost of gas, including demand and commodity charges, dollars per thousand cubic feet.
C_g = calorific value of gas, Btu per cubic foot.
E_g = over-all or house efficiency for gas, expressed as a decimal.

Example 6. If manufactured gas, having a calorific value of 535 Btu per cubic foot, costs 60¢ per thousand cubic feet, the cost per million Btu, assuming an efficiency of 80 per cent, will be:

$$Z = \frac{1000 \times 0.60}{535 \times 0.80} = \$1.40$$

Chapter 30

RADIATORS AND GRAVITY CONVECTORS

Heat Emission of Radiators and Convectors, Types of Radiators, Output of Radiators, Heating Effect, Heating Up the Radiator, Enclosed Radiators, Convectors, Code Tests, Gravity-Indirect Heating Systems

COMMERCIAL heating units are termed (1) *radiator*, for direct surfaces, either exposed, enclosed, or shielded; and (2) *convector*, or concealed heater, for extended surfaces that are built in as part of an enclosure or cabinet. Some heating units are also available that are a combination of radiator and convector.

HEAT EMISSION OF RADIATORS AND CONVECTORS

All heating units emit heat by *radiation* and *conduction*. The resultant heat from these processes depends upon whether or not the heating unit is exposed or enclosed and upon the contour and surface characteristics of the material in the units.

An exposed radiator emits less than half of its heat by radiation, the amount depending upon the size and number of sections. When the radiator is enclosed or shielded, radiation is further reduced. The balance of the emission is by conduction to the air in contact with the heating surface, and the resulting circulation of the air warms by convection.

A built-in heating unit in a convector emits practically all of its heat by conduction to the air surrounding it and this heated air is in turn transmitted by convection to the rooms or spaces to be warmed, the heat emitted by radiation being negligible. The small amount of heat transmitted by radiation to the inside surface of the enclosure diminishes as the surface temperature of the enclosure approaches the surface temperature of the heating unit.

TYPES OF RADIATORS

Present day radiators may be classified as tubular, wall, or window types, and are generally made of cast iron. Catalogs showing the many designs and patterns available now include a junior size which is more compact than the standard unit.

Pipe Coil Radiators

Pipe coils are assemblies of standard pipe or tubing (1 in. to 2 in.) which are used as radiators. In older practice these coils were commonly used in factory buildings, but now wall type radiators are most frequently used

for this service. When coils are used, the miter type assembly is to be preferred as it best cares for expansion in the pipe. Cast manifolds or headers, known as branch tees, are available for this construction.

OUTPUT OF RADIATORS

The output of a radiator can be measured only by the heat it emits. The old standard of comparison used to be square feet of *actual surface*, but since the advance in radiator design and proportions, the surface area alone is not a true index of output. (The engineering unit of output is now the *Mb* or 1000 Btu). However, during the period of transition from the old to the new, radiators may be referred to in terms of *equivalent square feet*. For steam service this is based on an emission of 240 Btu per hour per square foot.

TABLE 1. VARIATION IN DIMENSIONS AND CATALOG RATING OF 10-SECTION TUBULAR RADIATORS MADE BY SEVERAL MANUFACTURERS

No. of Tubes	3	4	5	6	7
Width of Radiator, Inches	4.6–5.1	6.0–7.0	8.0–8.9	9.1–10.4	11.4–12.8
Length per Section, Inches	2.5	2.5	2.5	2.5	2.5–3.0
HEIGHT WITH LEGS—INCHES	HEAT EMISSION—EQUIVALENT SQUARE FEET				
13–14	....	....	....	20	25.0–32.5
16–18	....	....	28.5	....	30.0–38.3
20–21	15.0–17.5	20.0–22.5	25.0–31.2	30	36.7–45.0
22–23	20.0–21.3	25	30.0–33.9	35	40.0–45.2
25–26	20.0–26.7	25.0–27.5	32.5–39.8	37.5–40.0	50.0–53.5
30–32	25.0–30.9	33.3–35.0	40.0–48.6	50	63.3–62.5
36–38	30.0–36.7	40.0–42.5	50.0–56.5	60	70.0–75.4

Output of Tubular Radiators

Table 1 illustrates the difficulty in tabulating tubular radiator outputs since there is so much variation between the products of the different manufacturers. Only on the four-tube and six-tube sizes is there any practical agreement in output value. The heat emission values appear as square feet but are entirely empirical, being based on the heat emission of the radiator and not on the measured surface.

Output of Wall Radiators

An average value of 300 Btu per actual square foot of surface area per hour has been found for wall radiators one section high placed with their bars vertical. Several recent tests[1] show that this value will be reduced from 5 to 10 per cent if the radiator is placed near the ceiling with the bars horizontal and in an air temperature exceeding 70 F. When radiators are placed near the ceiling, there is usually so noticeable a difference in temperature between the floor level and the ceiling that it becomes difficult to heat the living zone of a room satisfactorily.

[1]University of Illinois, *Engineering Experiment Station Bulletin No.* 223, p. 30.

Output of Pipe Coils

The heat emission of pipe coils placed vertically on a wall with the pipes horizontal is given in Table 2. This has been developed from available data and does not represent definite results of tests. For such coils the heat emission varies as the height of the coil. It is customary to use an average emission of 100 Btu per linear foot of 1¼-in. pipe, 10 ft high. The heat emission of each pipe of ceiling coils, placed horizontally, is about 126 Btu, 156 Btu, and 175 Btu per linear foot of pipe, respectively, for 1-in., 1¼-in., and 1½-in. coils.

TABLE 2. HEAT EMISSION OF PIPE COILS PLACED VERTICALLY ON A WALL (PIPES HORIZONTAL) CONTAINING STEAM AT 215 F AND SURROUNDED WITH AIR AT 70 F

Btu per linear foot of coil per hour (not linear feet of pipe)

SIZE OF PIPE	1 IN.	1¼ IN.	1½ IN
Single Row	132	162	185
Two	252	312	348
Four	440	545	616
Six	567	702	793
Eight	651	796	907
Ten	732	907	1020
Twelve	812	1005	1135

Effect of Paint

The prime coat of paint on a radiator has little effect on the heat output, but the finishing coat of paint does influence the radiation emission. Since this is a surface effect, there is no noticeable change in the convection loss. Thus, the larger the proportion of direct radiating surface, the greater will be the effect of painting on the radiation. Available tests are on old-style column type radiators which gave results shown in Table 3.

TABLE 3. EFFECT OF PAINTING 32-IN. THREE COLUMN, SIX-SECTION CAST-IRON RADIATOR[a]

RADIATOR No.	FINISH	AREA SQ FT	COEFFICIENT OF HEAT TRANS. BTU	RELATIVE HEATING VALUE PER CENT
1	Bare iron, foundry finish	27	1.77	100.5
2	One coat of aluminum bronze	27	1.60	90.8
3	Gray paint dipped	27	1.78	101.1
4	One coat dull black Pecora paint	27	1.76	100.0

[a]Comparative Tests of Radiator Finishes, by W. H. Severns (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927).

HEATING EFFECT

For several years the *heating effect* of radiators has been considered by engineers in order to use it for the rating of radiators and in the design of heating systems. Heating effect is the *useful output* of a radiator, in the comfort zone of a room, as related to the total input of the radiator[2].

[2]The Heating Effect of Radiators, by Dr. Charles Brabbée (A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 33).

The results of tests conducted at the University of Illinois are shown in Figs. 1 and 2[3]. For the four types of radiators shown, the following conclusions are given:

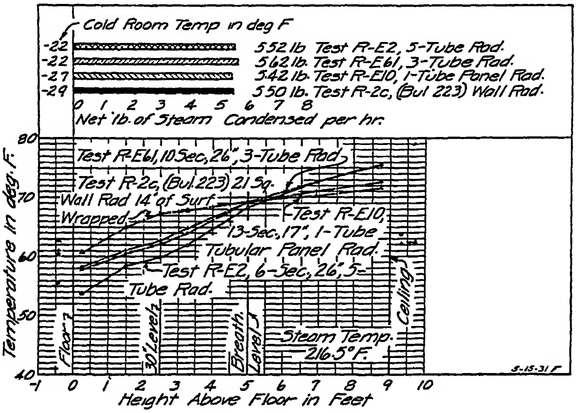

Fig. 1. Room Temperature Gradients and Steam Condensing Rates for Four Types of C. I. Radiators with a Common 60-in. Level Temperature

Note that the steam condensations are practically the same for all four radiators *when the same air temperature of 69 F is maintained at the 60-in. level.*

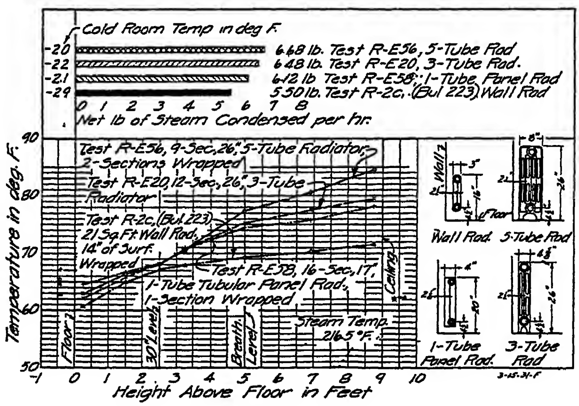

Fig. 2. Room Temperature Gradients and Steam Condensing Rates for Four Types of C. I. Radiators with a Common 30-in. Level Temperature

Note that the steam condensations are different for all four radiators *when the same air temperature of 68 F is maintained at the 30-in. level.*

1. The heating effect of a radiator cannot be judged solely by the amount of steam condensed within the radiator.

2. Smaller floor to ceiling temperature differentials can be maintained with long, low, thin, direct radiators, than is possible with high, direct radiators.

[3]Steam Condensation an Inverse Index of Heating Effect, by A. P. Kratz and M. K. Fahnestock (A.S.H.V.E. Transactions, Vol. 37, 1931).

3. The larger portion of the floor to ceiling temperature differential in a room of average ceiling height heated with direct radiators occurs between the floor and the breathing level.

4. The comfort level (approximately 2 ft-6 in. above floor) is below the breathing line level (approximately 5 ft-0 in. above floor), and temperatures taken at the breathing line may not be indicative of the actual heating effect of a radiator in the room. The comfort indicating temperature should be taken below the breathing line level.

5. High column radiators placed at the sides of window openings do not produce as comfortable heating effects as long, low, direct radiators placed beneath window openings[4].

HEATING UP THE RADIATOR

The maximum condensation occurs in a heating unit when the steam is first turned on. Fig. 3 shows a typical curve for the condensation rate in pounds per hour for the time elapsing after steam is turned into a cast-iron radiator. The data are from tests on old style column type radiators.

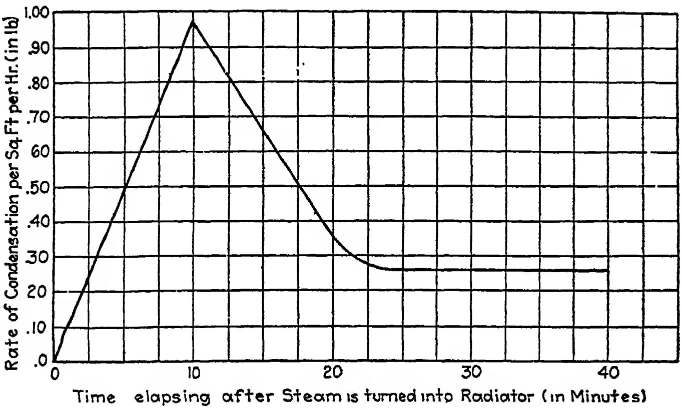

FIG. 3. CHART SHOWING THE STEAM DEMAND RATE FOR HEATING UP A CAST-IRON RADIATOR WITH FREE AIR VENTING AND AMPLE STEAM SUPPLY

In practice the rate of steam supply to the heating unit while heating up is frequently retarded by controlled elimination of air through air valves or traps. Automatic control valves may also retard the supply of steam.

ENCLOSED RADIATORS

The general effect of an enclosure placed about a direct radiator is to restrict the air flow, diminish the radiation and, when properly designed, improve the heating effect. Recent investigations[5] indicate that in the design of the enclosure three things should be considered:

1. There should be better distribution of the heat below the breathing line level to produce greater heating comfort and lowered ceiling temperatures.

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[4]Effect of Two Types of Cast Iron Steam Radiators in Room Heating, by A. C. Willard and M. K. Fahnestock (*Heating, Piping and Air Conditioning*, March, 1930).

[5]University of Illinois *Engineering Experiment Station Bulletins Nos.* 192 and 223, and Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, by A. C. Willard, A. P. Kratz, M. K. Fahnestock and S. Konzo (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

2. The lessened steam consumption may not materially change the radiator heating performance.

3. The enclosed radiator may inadequately heat the space.

A comparison between a bare or exposed radiator (*A*) and the same radiator with a well-designed enclosure (*B*), with a poorly-designed enclosure (*C*), and with a cloth cover (*D*) will illustrate the relative heating effects. In Fig. 4 the curve (*B*) reveals that the enclosed radiator used less steam than the exposed radiator, but gave a satisfactory heating performance. A well-designed shield placed over a radiator gives about the same heating effect. Curve (*C*) shows the unsatisfactory effects produced by improperly designed enclosures. Curve (*D*) shows that the effect of a cloth cover extending downward 6 in. from the top of the radiator was to make the performance unsatisfactory and inadequate.

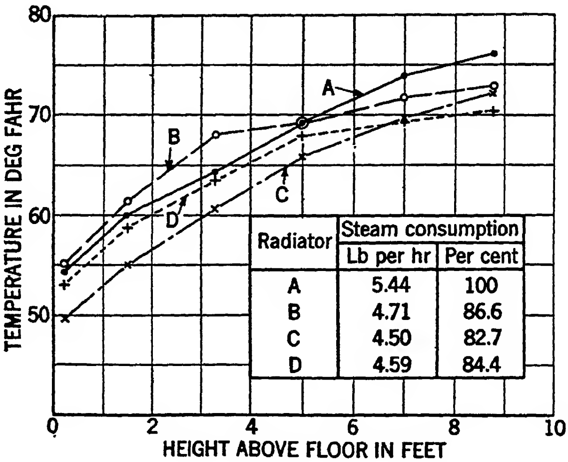
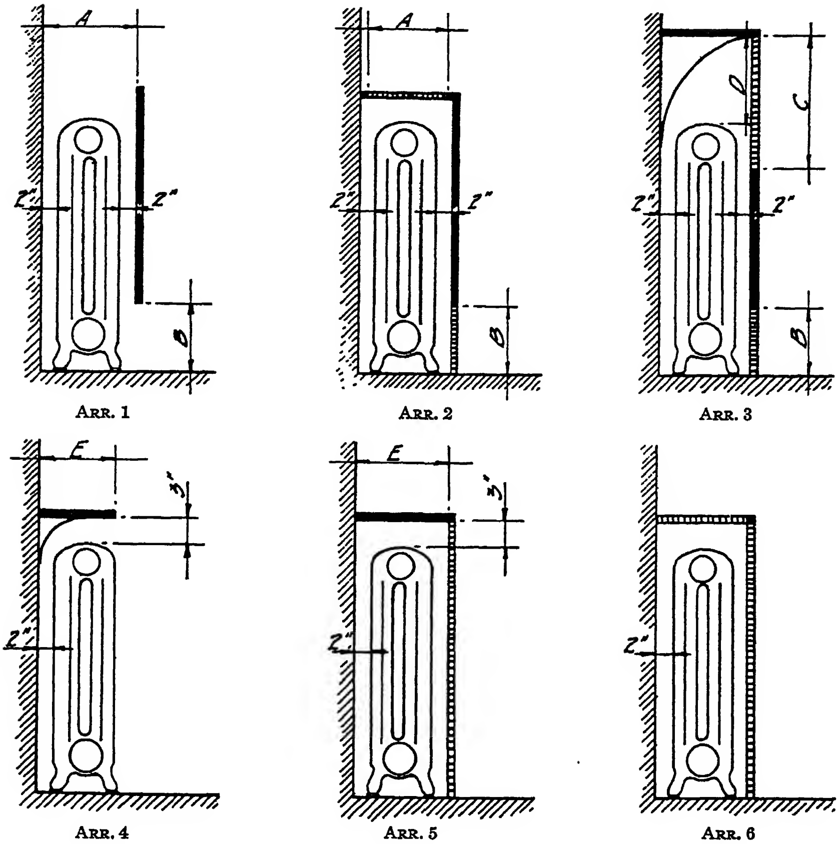


Fig. 4. Difference in Steam Pressure on Water in Boiler and at End of Steam Main

A practical interpretation of allowances to be made for radiator performance with various styles and forms of enclosures is shown by the diagrams and values listed in Fig. 5.

Practically all commercial enclosures and shields for use on direct radiators are equipped with water pans for the purpose of adding moisture to the air in the room. Tests[6] show that an average evaporative rate of about 0.235 lb per square foot of water surface per hour may be obtained from such pans, when the radiator is steam hot and the relative humidity in the room is between 25 and 40 per cent. This source of supply of moisture alone is not adequate to maintain a relative humidity above 25 per cent on a zero day.

[6]University of Illinois *Engineering Experiment Station Bulletin No.* 230, p. 20.



TYPE	INSTALLATION CONDITIONS	HEAT EMISSION SHALL BE ALTERED BY PERCENTAGE INDICATED
Arr. 1	When dimension A is as shown in Arr. 1 and dimension B is equal to 80 per cent of A	10% increase
Arr. 2	When dimension A is as shown and dimension B is equal to 80 per cent of A	5% increase
Arr. 3	When dimension B is equal to 80 per cent of A (as in Arr. 1), dimension C is equal to 150 per cent of A and dimension D is equal to A	No change
Arr. 4	When dimension E is equal to 50 per cent of A When dimension E is equal to A When dimension E is equal to 150 per cent of A	10% reduction 20% reduction 35% reduction
Arr. 5	When dimension E is equal to A	30% reduction
Arr. 6	When as shown	5% reduction

FIG. 5. TYPES OF ENCLOSURES AND HEAT EMISSION OF ENCLOSED RADIATORS[a]

[a]From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929).

CONVECTORS OR CONCEALED HEATERS

Although any standard heating unit (*i.e.*, radiator) may be concealed in a cabinet or other enclosure so that the greatest percentage of heat is conveyed to the room by convection, the best results are usually obtained where units of special design are used. Commercially, these specially designed units are built in as part of the enclosing cabinets which are necessary for the proper functioning of these heaters. As distinguished from *radiators*, these gravity convectors have come to be known as concealed heaters. Fig. 6 shows a typical built-in cabinet convector.

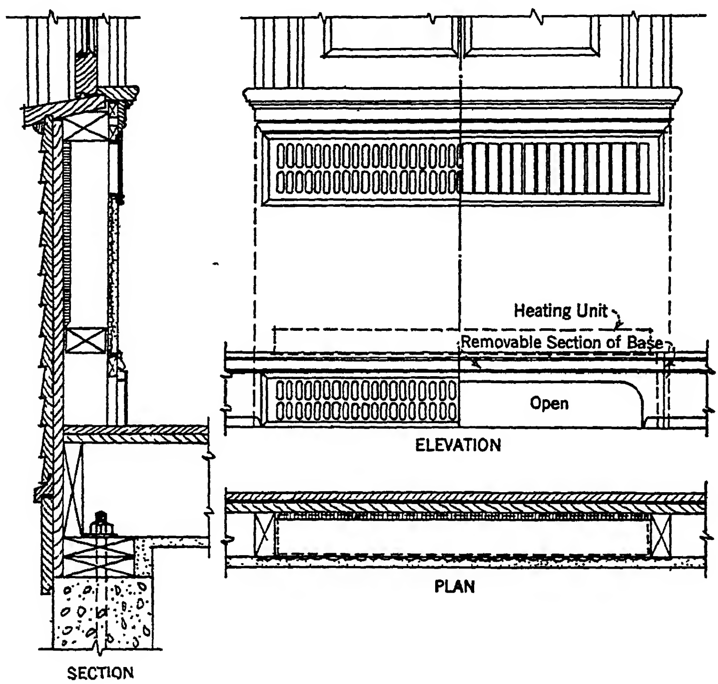


Fig. 6. Typical Concealed Convector Using Specially Designed Heating Unit

The elements or heating units usually consist of a relatively large amount of extended surface which may be integral with the core or assembled over it, making thermal contact by pressure, through solder, or by both pressure and metallic contact. Heating elements may be of cast-iron, cast aluminum, sheet steel, copper, or commercial alloys.

The cross-sections of a number of non-ferrous heating elements are shown in Fig. 7. In these the ratio between the extended surface and the prime surface may be as high as 25 to 1.

Concealed heaters or convectors maintain room temperatures with low steam consumption due, probably, to their performance characteristics which give reduced air temperatures in the upper level of a room with a

directed flow of warm air into the living zone and but little radiant heat to exposed surfaces. In listing the capacities of convectors, manufacturers allow from 10 to 40 per cent for this heating effect. (See A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation, A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931).

Concealed heaters or convectors are generally sold as completely built-in units. The enclosing cabinet should be designed with suitable air inlet and outlet grilles to give the heating element its best performance. Tables of capacities are catalogued for various lengths, depths and heights, and combinations are available in several styles for installations, such as the wall hanging type, free-standing floor type, recess type set flush with wall or offset, and the completely concealed type. Most of these types may be arranged with a top outlet grille, although the front outlet

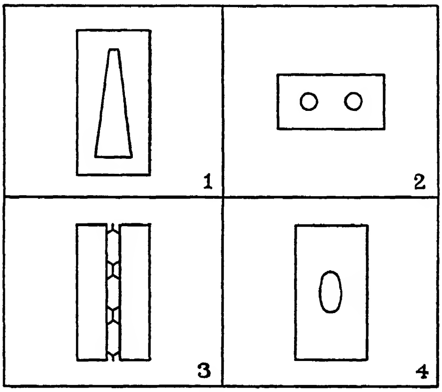


FIG. 7. SECTIONS THROUGH TYPICAL HEATING UNITS OF EXTENDED SURFACE TYPE

is practically standard. In cases where enclosures are to be used but are not furnished by the heater manufacturer, it is important that the proportions of the cabinet and the grilles be so designed that they will not impair the performance of the assembled convector.

The output of a concealed heater, for any given length and depth, is a variable of the height. Published ratings are generally given in terms of equivalent square feet, corrected for heating effect. However, an extended surface heating unit is entirely different structurally and physically from a direct radiator and, since it has no area measurement corresponding to the heating surface of a radiator, many engineers believe that the performance of convectors should be stated in Btu's. For steam convectors, as for radiators, 240 Btu per hour may be taken as an equivalent square foot of radiation.

CODE TESTS FOR RADIATORS AND CONVECTORS

As previously indicated, the output of radiators and convectors is still designated by the terms of older practice, but this is gradually giving place to an engineering method of designating heat emission. The A.S.H.V.E.

has adopted the following standards: Code for Testing Radiators (1927); Codes for Testing and Rating Concealed Gravity Type Radiation (Steam, 1932, and Hot Water, 1933).

For steam services the actual condensation weight is taken without any allowance for heating effect; for hot water services the weight of circulated water is used without allowance for heating effect. In all cases the total heat transmission varies as the 1.3 power of the temperature difference between that inside the radiator and the air in the room, and is expressed in Btu or *Mb* per hour.

Standard test conditions specify either a steam pressure of 1 lb gage (215 F), or hot water at 170 F and a room temperature of 70 F for radiators, or an inlet air temperature of 65 F for convectors. The heating capacity of a *steam radiator* or *steam convector* is determined as follows:

$$H_t = W_s h_{fg} \tag{1}$$

where

H_t = Btu per hour under test conditions.
W_s = condensation in lb per hour.
h_{fg} = latent heat in Btu per lb.

H_t may be converted to standard conditions of code ratings by using the proper correction factor from the following formulae:

For radiators:

$$C_s = \left(\frac{215 - 70}{T_s - T_r}\right)^{1.3} = \left(\frac{145}{T_s - T_r}\right)^{1.3} \tag{2}$$

For convectors:

$$C_s = \left(\frac{215 - 65}{T_s - T_i}\right)^{1.3} = \left(\frac{150}{T_s - T_i}\right)^{1.3} \tag{3}$$

The output under standard conditions will be:

$$H_s = C_s H_t \tag{4}$$

where

C_s = correction factor.
T_s = steam temperature during test in degrees Fahrenheit.
T_r = room temperature during test in degrees Fahrenheit.
T_i = inlet air temperature during test in degrees Fahrenheit.
H_s = heat emission rating under standard conditions in Btu per hour.

Similarly, for *hot water convectors*, the output under test conditions may be determined as follows:

$$H = W(\theta_1 - \theta_2)\frac{3600}{t} \tag{5}$$

where

H = Btu per hour under test conditions.
W = pounds of water handled during test.
θ_1 = average temperature of inlet water in degrees Fahrenheit.
θ_2 = average temperature of outlet water in degrees Fahrenheit.
t = duration of test in seconds.

To convert test results to standard conditions, the following correction factor is used:

$$C = \left(\frac{170 - 65}{\frac{\theta_1 + \theta_2}{2} - T_i}\right)^{1.3} = \left(\frac{105}{\frac{\theta_1 - \theta_2}{2} - T_i}\right)^{1.3} \tag{6}$$

It has been shown for the exponent 1.3 that the range of error when it is used is less than 5 per cent[7].

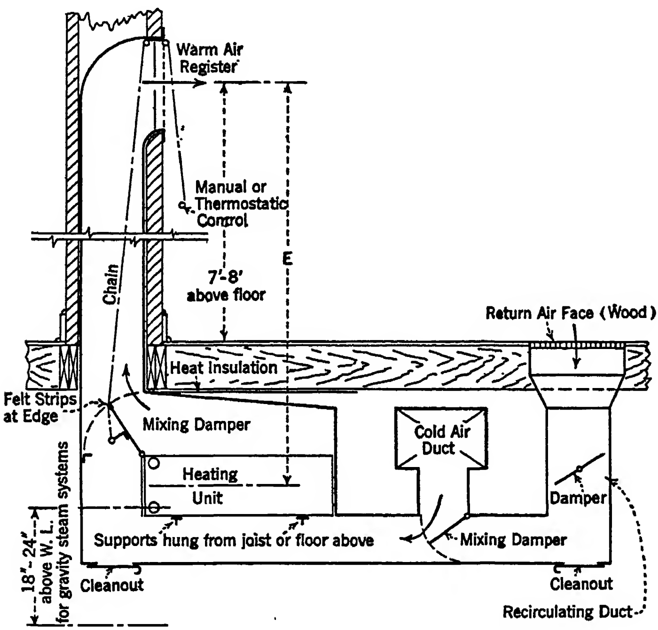


FIG. 8. GRAVITY-INDIRECT HEATING SYSTEM[a]

[a]See *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

GRAVITY-INDIRECT HEATING SYSTEMS[8]

The heating units for this system are usually of the extended surface type for steam or hot water, and are installed about as shown in Fig. 8. The temperature and volume of the air leaving the register must be great enough so that in cooling to room temperature the heat available will just equal the heat loss during the same time. In cases where ventilation is a requirement, the air volume needed may become so large that the entering

[7]Tests of Convectors in a Warm Wall Testing Booth, by A. P. Kratz, M. K. Fahnestock, and E. L. Broderick (*Heating, Piping* and *Air Conditioning*, August, 1933).

[8]For further information on this subject see A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings (edition of 1929) and *Mechanical Equipment of Buildings*, by Harding and Willard, Vol. I, second edition, 1929.

air temperature will be but slightly above the room temperature. To establish and maintain a constant heat flow, provision must be made for removing the air in the room, after it has cooled to the desired room temperature, by a system of vent flues or ducts. As the air flow is maintained by natural draft and this gravity head is very slight, it is necessary to make all ducts as short as possible, especially the runs from the heating units to the base of the vertical warm air flues.

Chapter 31

STEAM HEATING SYSTEMS

Gravity and Mechanical Return, Gravity One-Pipe Air-Vent System, Gravity Two-Pipe Air-Vent System, One-Pipe Vapor System, Two-Pipe Vapor System, Atmospheric System, Vacuum System, Sub-Atmospheric System, Orifice System, Zone Control, Condensation Return Pumps, Vacuum Pumps, Traps

STEAM heating systems may be classified according to the piping arrangement, the accessories used, the method of returning the condensate to the boiler, the method of expelling air from the system, or the type of control employed. The essential features of the common types are described in this chapter. Information concerning the design and layout of steam heating systems is given in Chapter 32.

GRAVITY AND MECHANICAL RETURN

In *gravity systems* the condensate is returned to the boiler by gravity due to the static head of water in the return mains. The elevation of the boiler water line must consequently be sufficiently below the lowest heating units and steam main and dry return mains to permit the return of condensate by gravity. The *water line difference** must be sufficient to overcome the maximum pressure drop in the system and, when radiator and drip traps are used as in two-pipe vapor systems, the operating pressure of the boiler. This applies only to closed circuit systems, where the condensation is returned to the boiler. If the condensation is wasted, no water line difference is required.

In *mechanical systems* the condensate flows to a receiver and is then forced into the boiler against the boiler pressure. The lowest parts of the supply side of the system must be kept sufficiently above the water line of the receiver to insure adequate drainage of water from the system, but the relative elevation of the boiler water line is unimportant in such cases except that the head on the pump or trap discharge becomes greater as the height of the boiler water line above the trap or pump increases.

There are three general types of mechanical returns in common use, namely, (1) the mechanical return trap, (2) the condensation return pump, and (3) the vacuum return pump. Further information on pumps and traps will be presented later in this chapter.

GRAVITY ONE-PIPE AIR-VENT SYSTEM

In the gravity one-pipe air-vent system each radiator has but a single connection through which steam must enter and condensation must

*The *water line difference* is the distance between the water line of the boiler and the low point of the water in the dry return main.

return in the opposite direction. Each radiator has an individual air valve.

Up-Feed Gravity One-Pipe Air-Vent System

This system is the most common of all methods of steam heating, due largely to its low cost of installation and its simplicity. As will be seen from Fig. 1, the steam piping rises to a point as high as possible at the boiler and pitches downward from this location until the far end of the main or mains is reached. At the far ends drips are taken off at the low points of the steam mains, are water-sealed below the boiler water line, and then brought back to the boiler in a wet return. Single pipe risers

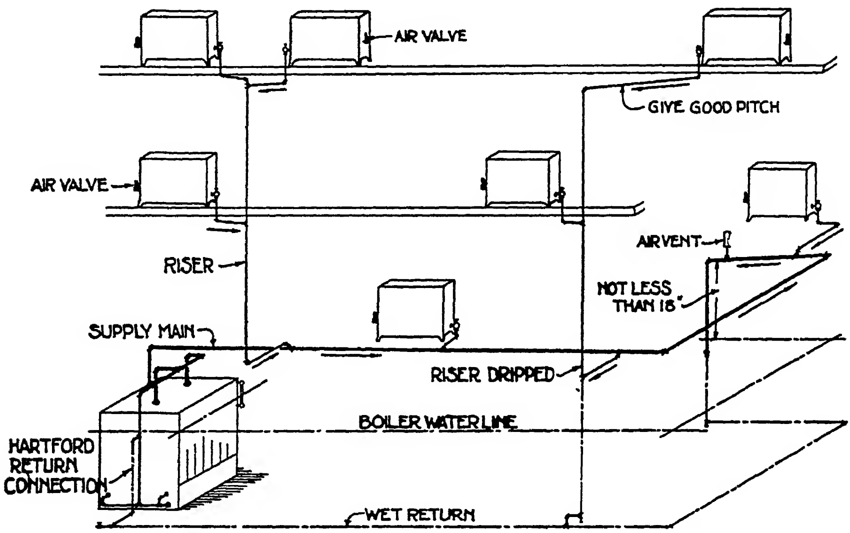


FIG. 1. TYPICAL UP-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

are branched off the main or mains to feed the radiators, the steam passing up the riser and the condensation flowing down it. The steam and condensation flow in opposite directions in the riser but after the condensation enters the steam main it flows in the same direction as the steam and is disposed of through the drip connection at the end of the main. In buildings of several stories, it is customary to drip the heel of each riser separately, whereas in one- or two-story buildings this is not necessary. Both types of branches and risers are shown in Fig. 1.

Horizontal branches to radiators and risers should be pitched at least ½-in. in 10 ft downward toward the riser or vertical pipe, and the horizontal branches from the steam main should be graded at least this amount toward the main, except where the heel of the riser is dripped, in which case the branch should pitch down toward the riser drip (Figs. 2 and 3). The return line, if wet, may be run without pitch or may be pitched in either direction, but if it is necessary to carry the return main

overhead for any distance before dropping, the return should slope downward with the flow.

The radiator valves may be of the angle-globe or gate type. They should not be of the straight-globe type because the damming effect of the raised valve seat interferes with the flow of condensation through the valve. Graduated valves cannot be used, as the steam valves on this system must be fully open or closed to prevent the radiators' filling with water. Air valves may be manual or automatic, with or without a check to prevent the re-entrance of expelled air. Usually the automatic type is installed. The greatest source of difficulty with one-pipe steam systems is that the heat is all on or all off, with no intermediate position possible. However, intelligent use of the on-and-off method of manual control gives reasonably satisfactory results.

It is important that the lowest points of the steam mains and heating units be kept sufficiently above the water line of the boiler to prevent

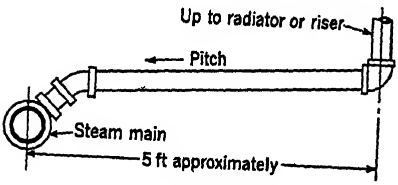


FIG. 2. TYPICAL STEAM RUNOUT WHERE RISERS ARE NOT DRIPPED

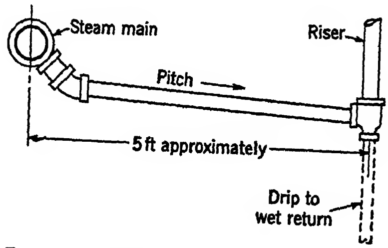


FIG. 3. TYPICAL STEAM RUNOUT WHERE RISERS ARE DRIPPED

flooding, although proper design will eliminate this danger. Usually 18 in. is sufficient but construction limitations frequently make shorter distances necessary. The distance may be checked in the following manner:

Referring to Fig. 4 it will be seen that the water in the wet return is really in an inverted siphon, or U-shaped container, with the boiler steam pressure on the top of the water at one end and the steam main pressure on the top of the water at the other end. The difference between these two pressures is the *pressure drop* in the system, *i.e.*, the friction of the steam in passing from the boiler to the far end of the main. The water in the far end will rise sufficiently to overcome this difference in order to balance the pressures, and it will rise enough farther to produce a flow through the return into the boiler (usually about 3 in. unless the pipes are small or full of sediment), and it will rise still farther if a check valve is installed in the return so as to obtain sufficient head to lift the tongue of the check (usually 4 in. will be necessary).

If a one-pipe steam system is designed, for example, for a total pressure drop of ⅛ lb, and utilizes an Underwriters Loop[1] instead of a check valve on the return, the rise in the water level at the far end of the return due to the difference in steam pressure would be ⅛ of 28 in., or 3½ in. Adding 3 in. to this for the flow through the return main and 6 in. as a factor of safety gives 12½ in. as the distance the bottom of the lowest part of the steam main and all heating units must be above the boiler water line. The same system, however, installed and sized for a total pressure drop of ½ lb, and with a check in the return, would require ½ of 28 in., or 14 in., for the difference in steam pressure, 3 in. for the flow through the return, 4 in. to operate the check, and 6 in. for a factor of safety, making a total of 27 in. as the required distance. Higher pressure drops would increase the distance accordingly.

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[1]See discussion of piping details in Chapter 32.

Down-Feed Gravity One-Pipe Air-Vent System

In the overhead down-feed gravity one-pipe air-vent system there is no change over the *up-feed system* in the radiators, the radiator valves, the air valves, or the radiator runouts as far back as the risers. Beyond this point there are basic differences. The steam is taken from the boiler and carried to the top of the building as near the boiler as possible (Fig. 5). If the run to the main riser is long, or if the riser extends several stories in order to reach the top, the bottom of the riser should be dripped into the wet return. The horizontal main is taken off the top of the riser and grades down from the riser toward all of the drops, each drop taking its share of the main condensation (Fig. 6), or all of the drops except the last may be taken from the top of the main (Fig. 7), the last drop being from the bottom and serving as a drain for the entire main. As the overhead

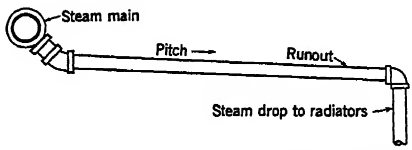


FIG. 6. STEAM RUNOUTS DRIPPING MAIN

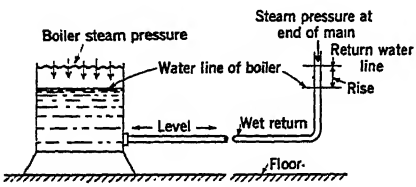


FIG. 4. DIFFERENCE IN STEAM PRESSURE ON WATER IN BOILER AND AT END OF STEAM MAIN

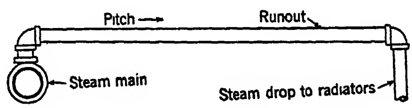


FIG. 7. STEAM RUNOUTS WITH MAIN DRIPPED AT END ONLY

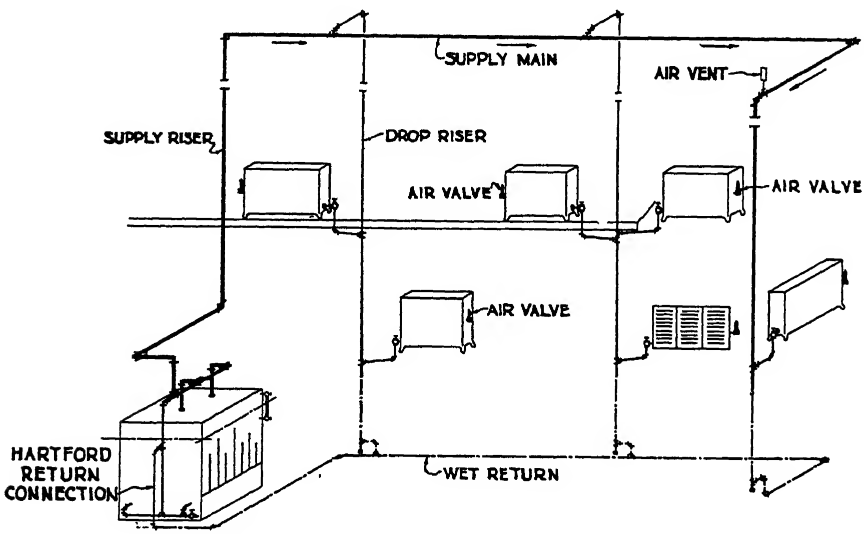


FIG. 5. TYPICAL DOWN-FEED GRAVITY ONE-PIPE AIR-VENT SYSTEM

main does not carry any condensation from the radiators it is immaterial which method is used. The air vent shown on the main just before the last drop (Fig. 5) may be placed at this point or it may be located at the bottom of the drop under the last radiator connection and sufficiently above the water line of the boiler to prevent flooding.

GRAVITY TWO-PIPE AIR-VENT SYSTEM

The gravity two-pipe system is now considered obsolete although many of these systems are still in use in older buildings. Separate supply and

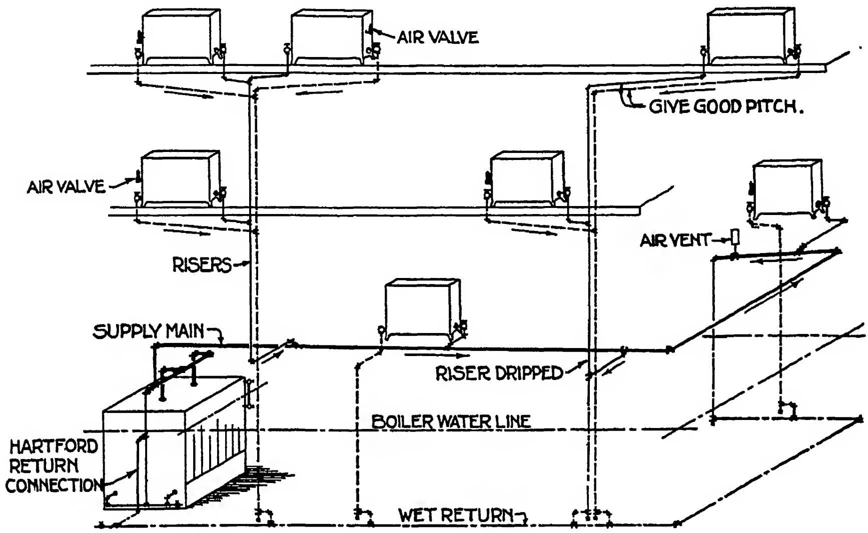


FIG. 8. TYPICAL UP-FEED GRAVITY TWO-PIPE AIR-VENT SYSTEM

return mains and connections are required for each heating unit; air valves are installed on the heating units and mains; hand valves are installed on the returns.

Up-Feed Gravity Two-Pipe System

This system (Fig. 8) has a steam and a return connection to each radiator. The radiator valves for steam, return, and air are the same as those described for the gravity one-pipe air-vent system. The steam main is run and pitched in the same manner as in the one-pipe system, but the returns from each radiator are connected into a separate return line system which has its risers carried down and joined to a wet return line under the boiler water line level. Where the return has to be kept high to function as a dry return, it is advisable to connect the return risers to the dry return main through water seals about 36 in. deep, as shown in Fig. 9, to prevent steam from one riser entering another and closing the air valves on the nearest radiators.

Down-Feed Gravity Two-Pipe System

The steam main in the down-feed system is carried to the top of the building, and the piping of the steam side is arranged practically as in the down-feed one-pipe gravity system. The drips at the bottoms of the steam drops and the runouts to the radiators are similar to those shown in Fig. 8 for the up-feed gravity two-pipe system. On the return side of the system, the piping is arranged in exactly the same manner as the up-feed gravity two-pipe system.

ONE-PIPE VAPOR SYSTEM

A vapor system is one which operates under pressures at or near atmospheric and which returns the condensation to the boiler by gravity. The piping arrangement of a one-pipe vapor system is similar to that of

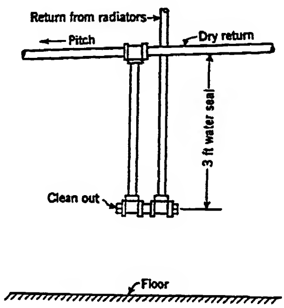


FIG. 9. METHOD OF CONNECTING TWO-PIPE GRAVITY RETURNS TO DRY RETURN MAIN

the gravity one-pipe steam system; in fact, one-pipe gravity installations may readily be changed to one-pipe vapor systems by making a few simple alterations. The steam radiator valve is a plug cock which when opened gives a free and unobstructed passageway for water. The automatic air valve is of special design to permit the ready release of air from the radiator and to prevent the return of the air after it is expelled. The air valves on the main are a quick relief type, and the whole system is designed to operate on a few ounces of pressure.

TWO-PIPE VAPOR SYSTEM

Two-pipe vapor systems may be classified as (1) *closed systems* consisting of those which have a device to prevent the return of air after it is once expelled from the system, and which can operate at sub-atmospheric pressures for a period of four to eight hours depending upon the tightness of the system, and (2) *open systems* consisting of those which have the return line constantly open to the atmosphere without a check or other device to prevent the return of air, and which operate at a few ounces above atmospheric pressure.

Under the first classification the essentials are packless graduated valves on the radiators, thermostatic return traps on the returns, and traps on all drips unless they are water sealed. Such a system, illustrated in Fig. 10, should be equipped with an automatic return trap to prevent the water from backing out of the boiler. In this up-feed arrangement the supply piping is carried to a high point directly at the boiler and is graded down toward the end or ends of the supply main, each supply main being dripped at the end into the wet return or carried back to a point near the boiler where it drops down below the boiler water line and becomes a wet return. From this main, runouts are branched off to feed risers or radiators above, these being graded back toward the steam main

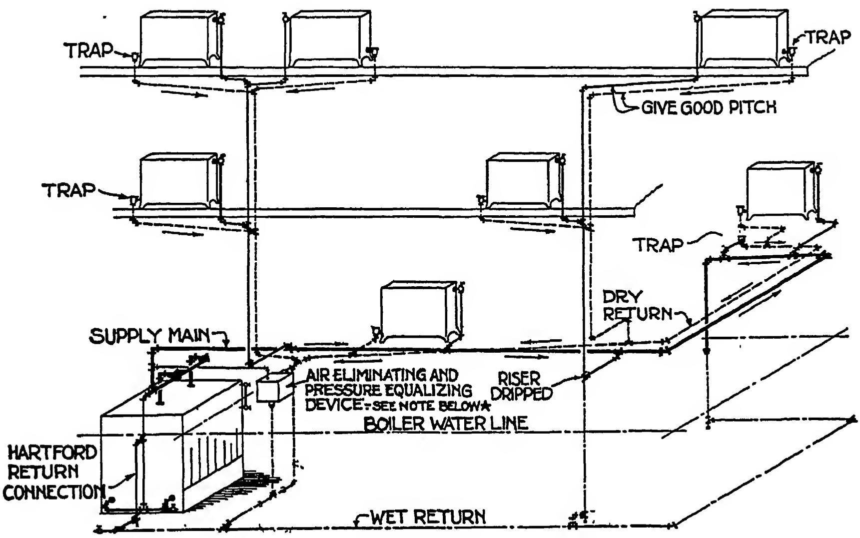


FIG. 10. TYPICAL UP-FEED VAPOR SYSTEM WITH AUTOMATIC RETURN TRAP[a]

[a]Proper piping connections are essential with special appliances for pressure equalizing and air elimination.

if they are not dripped at the bottom of the riser, or toward the riser if the riser heel is dripped. Both conditions are illustrated in Figs. 2 and 3.

Return risers are connected to each radiator on its return end through thermostatic traps. Their bottoms are connected to the return main through runouts which slope toward the main. The return main itself is sloped back toward the boiler if it is carried overhead; if run wet, the slope may be neglected. An air vent is installed at the point at which the return main drops below the water line. In the simplest cases this vent consists of a ¾-in. pipe with a check valve opening outward, but in certain patented systems special forms of vent valves, designed to allow the air readily to pass out of the system and to prevent its return, are used. A check valve is inserted in the return main at a point near the boiler and a vertical pipe is run up into the bottom of the return trap, which usually is located with the bottom about 18 in. above the boiler water line. Some traps are constructed to permit the bottom's being

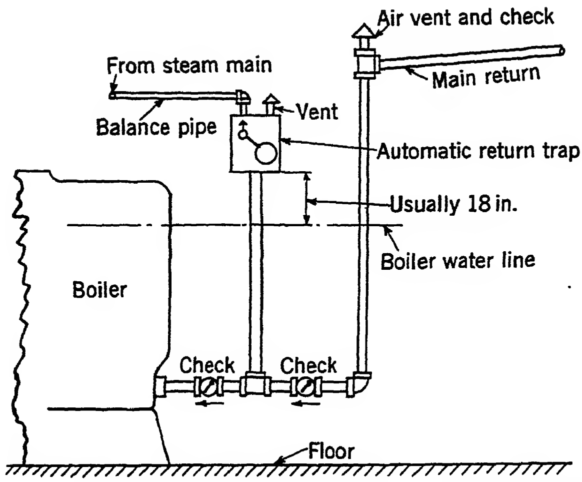


Fig. 11. Typical Connections for Automatic Return Trap

placed as close as 8 in. above the boiler water line. On the other side of this connection a second check valve is installed in the main return just before it enters the boiler. (Fig. 11).

Down-Feed Two-Pipe Vapor System

In the down-feed two-pipe vapor system the steam is carried to the top of the building, the top of the vertical riser constituting the high point of the system, and the horizontal supply main is sloped down from this location to the far ends of each branch. The branches are taken off the main from the bottom or at a 45 deg angle downward, with the runouts

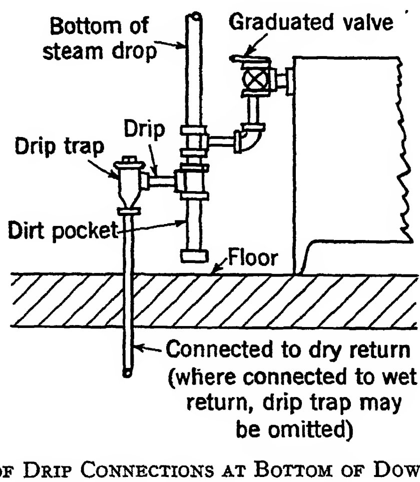


Fig. 12. Detail of Drip Connections at Bottom of Down-Feed Steam Drop

sloped toward the drops (Fig. 6). Thus each branch from the main forms a drip and no accumulation of water is carried down any one drop. Another method of running the steam main, which is not considered as satisfactory but which is practical, is to take the branches off the top of the main (Fig. 7) and to drip the end of the main through the last riser, as illustrated in the down-feed one-pipe system detail shown in Fig. 6. If this is done, the pipe drop at the end or ends of the mains should be enlarged one pipe size to provide capacity for this concentration of the main drip.

The steam drops are carried down through the building with suitable reductions as the various radiator connections are taken off until the lowest radiator runout is reached. If the drop is only two or three stories high, the portion feeding the bottom radiator should be increased one pipe size to provide for draining the riser, and if the drop is over three stories high it is well to increase the portion feeding the two lowest radiators one or two pipe sizes, especially if the two lowest radiators are small and the normal size of drop required is 1 in. or less. The bottom of the steam drops should terminate with a dirt pocket above which a drip trap connection is located, as shown in Fig. 12. The returns on a down-feed vapor system are the same as on an up-feed system except that every steam drop must have a drip at the bottom connected either into the return through a trap or into a separate water-sealed drip line below the boiler water line, as illustrated in Fig. 10, in which case the thermostatic traps may be omitted. The runouts to the radiators and the radiator connections of the down-feed system are the same as those of the up-feed system already described.

ATMOSPHERIC SYSTEM

The distinguishing features of the atmospheric system are gravity return to the boiler or to waste, graduated or ordinary radiator valves, no automatic air valves on the radiators, thermostatic traps on the radiator returns, and the venting of all air from the system by means of pipes open to the atmosphere. The returns are open to the atmosphere at all times, usually by extending the return risers to the top of the building where they are either connected together in groups and carried through the roof or extended through the roof individually. Atmospheric systems, either up-feed or down-feed, are often used where the condensation is not returned to the boiler, as in heating systems supplied by high pressure steam through pressure-reducing valves at locations far from the boilers. The returns may be delivered back to the boiler, if desired, by condensation return pumps which are vented to the atmosphere. The return lines in such systems are simply gravity waste lines in which the condensation flows entirely by gravity and is not aided by any pressure difference.

The steam side may be run as that for either up-feed or down-feed two-pipe vapor systems, as the conditions require, and the radiator connections are the same as for vapor systems in that they have graduated valves on the radiator supply ends and thermostatic traps on the radiator return ends. All drips from the supply main and the steam side of the system must pass through thermostatic drip traps before entering the return system where only atmospheric pressure exists. Fig. 13 illustrates a typical scheme of piping used on atmospheric systems.

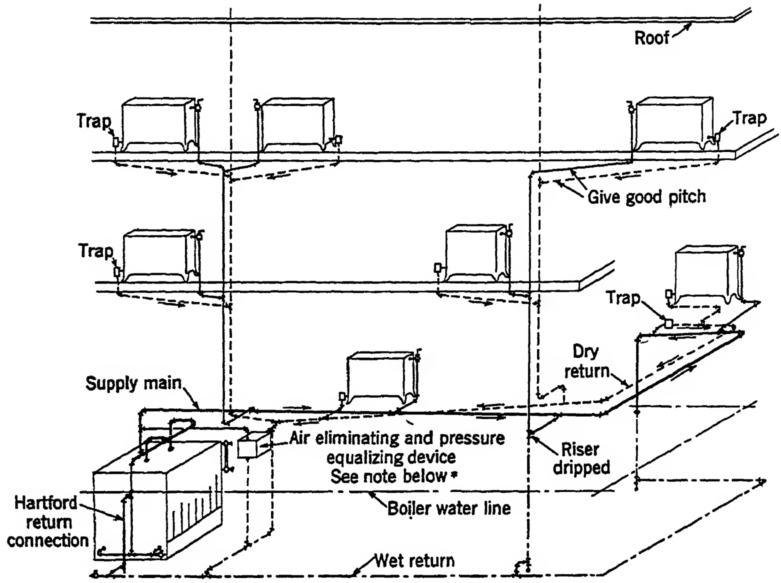


FIG. 13. TYPICAL ATMOSPHERIC SYSTEM WITH AUTOMATIC RETURN TRAP

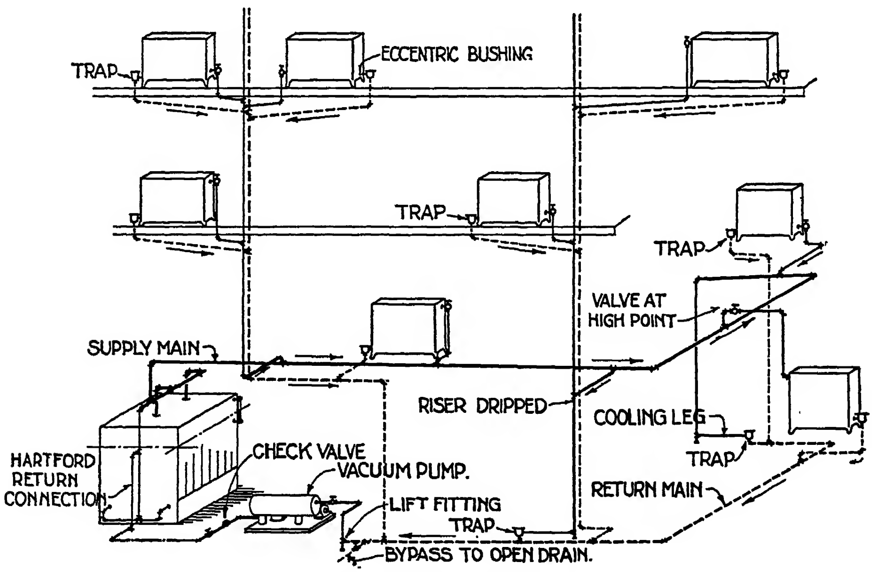


FIG. 14. TYPICAL UP-FEED VACUUM PUMP SYSTEM

VACUUM SYSTEM

In the vacuum system, a vacuum is maintained in the return line practically at all times but no vacuum is carried on the steam side, and the usual accessories include graduated valves on the radiator supply and thermostatic traps on the radiator return. The air is expelled from the system by a vacuum pump and all drips must pass through thermostatic traps before connecting to the return side of the system.

These systems are often fed from high pressure steam mains through pressure-reducing valves but they may be fed direct from a low-pressure steam heating boiler as shown in Fig. 14, in which a typical up-feed vacuum system is illustrated. The supply main slopes down in the direction of flow; the runouts pitch down toward the riser if the riser is dripped (Fig. 3) or up toward the riser if the riser is not dripped (Fig. 2); both conditions are indicated in Fig. 14. The matter of dripping the risers depends largely on the height of the riser and the judgment of the designer. Ordinarily risers less than three stories high are not dripped and those more than four stories high are dripped, but there is no set rule for this. When risers are dripped the runouts from the steam main may be taken from the bottom if desired and each runout then serves as a drip for the main.

The risers are carried up to the highest radiator connection and are connected to the radiator through runouts sloping back toward the riser. The radiators usually have graduated valves on the supply end, although this is not absolutely necessary. Angle-globe valves and gate valves may be used where graduated manual control is not desirable. The return valves must be of the thermostatic type which will pass air and water but which will close against the passage of steam.

The return risers are carried down to the basement and are connected into a common return line, care being taken that no air pockets exist in the runouts or in the horizontal return main which slopes downward toward the vacuum pump to which it is connected. The air and water are taken by the vacuum pump, which discharges the air from the system and pumps the water back to the boiler, or other receiver, which may be a feed-water tank or a hot well. It is essential on these systems that no connection from the supply side to the return side be made at any point except through a trap.

While the best practice demands a return flowing to the vacuum pump in an interrupted downward slope, in some cases limitations make it necessary to drop the return below the level of the vacuum pump inlet before the pump can be reached. In such event one of the advantages of the vacuum system is that the return can be raised by the suction of the vacuum pump for a considerable height, depending on the amount of vacuum maintained, by means of a lift fitting inserted in the return. When the lift is considerable, several lift fittings are used in steps (Fig. 15), more successful operation being obtained by this method than when the lift is made in one step. If the lift occurs close to the vacuum pump, a special arrangement is used as shown in Fig. 16.

Down-Feed Vacuum System

The piping arrangement for the down-feed vacuum system is similar on the supply side to the down-feed vapor system in that it has similar

runouts, radiator valves, drips on the bottom of the steam drops, and enlargement of the drops for the lower radiator connections. The return side of the system is exactly the same as the up-feed system except that the steam riser drips at the bottom are connected into the return line through thermostatic traps. It is preferable to take the runouts for the risers from the bottom or at a 45 deg angle down from the steam main (Fig. 6) so that they may serve as steam main drips. When this is done it is practical to run the steam main level if a runout is located at every change in pipe size, or if eccentric fittings are used (Fig. 17). A slight pitch in the steam main, however, should be used when possible. An overhead vacuum down-feed system is shown diagrammatically in Fig. 18.

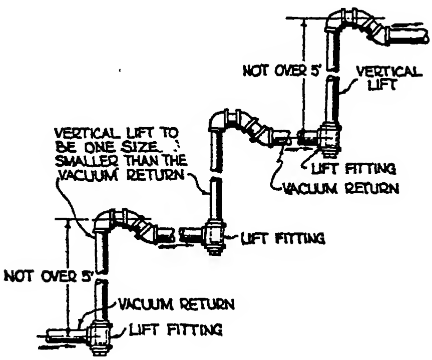


FIG. 15. METHOD OF MAKING LIFTS ON VACUUM SYSTEMS WHEN DISTANCE IS OVER 5 FT

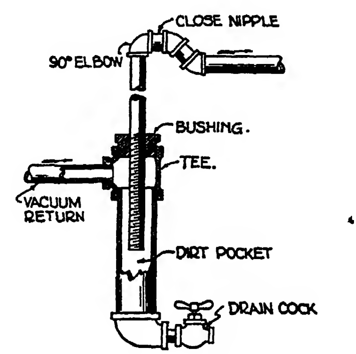


FIG. 16. DETAIL OF MAIN RETURN LIFT AT VACUUM PUMP

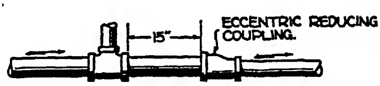


FIG. 17. METHOD OF CHANGING SIZE OF STEAM MAIN WHEN RUNOUTS ARE TAKEN FROM TOP

SUB-ATMOSPHERIC SYSTEMS

The sub-atmospheric systems are similar to the vacuum system except that a pump capable of operating up to 25 in. of vacuum is used, and a control is placed on the pump so that the vacuum or absolute pressure carried in the return can be maintained a certain amount below that existing in the steam line to cause a constant circulation. The traps are designed to operate in high vacuum. It is apparent that this system differs from the ordinary vacuum system by having a vacuum on both sides of the system, instead of only on the return side, in order to secure control of the heat emission from the radiators and thus to control the temperature in the building. The system can be operated in the same manner as the ordinary vacuum system when desired.

In the vacuum system, steam pressure above that of the atmosphere exists in the supply mains and radiators practically at all times. In the sub-atmospheric system, steam pressure exists in the steam main and

radiators only during the most severe weather, while under average winter temperatures the steam is under a partial vacuum which in mild weather may reach as high as 25 in. This vacuum is largely self-induced by the condensation of the steam in the system when an inadequate supply of steam is being furnished through the control valve which admits it. In the sub-atmospheric system, a control valve is inserted on the steam main of an ordinary vacuum system near the boiler, a high-vacuum pump is substituted for the ordinary type and is supplied with a pressure-difference control, and traps are placed on the radiators and drips which will operate satisfactorily at any pressure from 5 lb gage to 26 in. of vacuum.

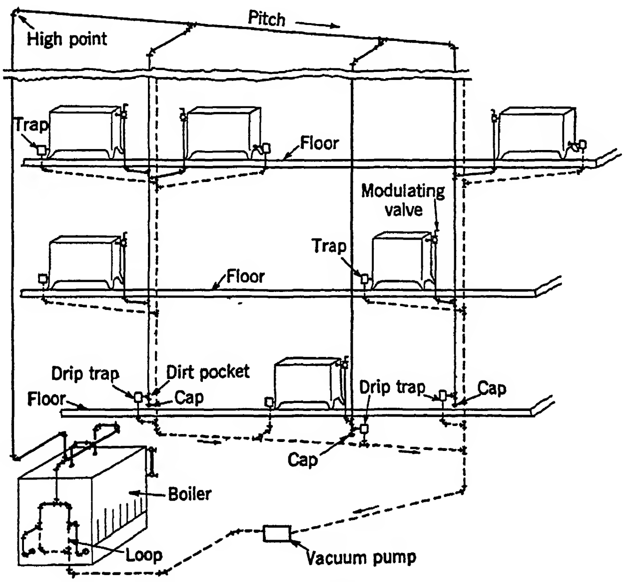


FIG. 18. TYPICAL DOWN-FEED VACUUM SYSTEM

The control valve is a special pressure reducing valve which may be controlled manually or thermostatically from points selected in the building. The vacuum pump regulator is simply a diaphragm so arranged that, when the vacuum in the return line is insufficient to hold the desired difference in pressure between the steam and return sides of the system, the vacuum pump is automatically started and the vacuum increased to the necessary amount. The actual pressure difference maintained between the two sides of the system is only enough to secure adequate circulation and is often about 2 in. of mercury. This fixed pressure difference between the supply and return sides of the system results in practically constant circulation under all pressure conditions.

In order to distribute the steam equally when the system is being warmed up and also to reduce the amount of steam delivered to the radiators on mild days, orifice plates are used in the graduated radiator control valves. The heat emitted from the radiators in mild weather and

under conditions of high vacuum is not only reduced in proportion to the difference in the steam temperature between that for 2 lb gage and for 25 in. of vacuum but it is reduced still further by a reduction in the amount of steam which can pass through the orifice when the steam is expanded due to the vacuum. This renders possible the control of heat emission from the radiators to a point not indicated entirely by the difference in steam temperatures.

The high-vacuum pumps on this system are equipped with receivers having float control so that the pump can be placed on a receiver-return-pump basis at night if desired so no high vacuum will be carried. One radical difference between this system and the ordinary vacuum system is that no lifts can be made in the return line. The returns must grade downward constantly and uninterruptedly from the radiator return outlet to the inlet on the high-vacuum pump receiver. No attempt should be made to heat service water on this system unless the steam line for water heating is taken off the boiler header back of the heating system control valve, and then only when 2 lb or more will be carried on the boiler at all times.

ORIFICE SYSTEM

Orifice systems of steam heating may have piping arrangements identical with vacuum systems but some of these systems omit both the radiator thermostatic traps and the vacuum pump in cases where the returns are wasted to a sewer or delivered to some type of receiver in which no back pressure exists. The principle on which they operate is embodied in the well-known fact that an orifice will deliver varying velocities when the ratio of the absolute pressures on the two sides of the orifice exceeds 58 per cent. If the absolute pressure on the outlet side is less than 58 per cent of the absolute pressure on the inlet side no further increase in velocity will be obtained.

As a result, if an orifice is so designed in size as to exactly fill a radiator with steam at 2 lb gage on one side and ¼ lb gage on the other, the absolute pressure relation is

$$\frac{14.7 + 0.25}{14.7 + 2.0} = 90 \text{ per cent}$$

Should the steam pressure be dropped to ¼ lb gage, the pressure on each side of the orifice would be balanced and no steam flow would take place. From this it will be seen that if an orifice of a given diameter will fill a given radiator with steam when there is a given pressure on the main, it is simply a question of dropping this main pressure so as to fill any desired portion of the radiator down to the point where the main pressure equals the back pressure in the radiator, at which time no steam will be supplied at all. If orifices throughout a job are designed on a similar basis, all radiators will heat proportionately to the steam pressure within the limits for which the orifices are designed.

Some systems use orifices not only in radiator inlets but also at different points on the main, thus balancing the system to a greater extent. For example, the system may be designed for a particularly long run involving an initial pressure of 3 lb gage on the main and 2 lb at the end of the main,

but each branch from the main may have an orifice for reducing the pressure at it to 2 lb gage. This is particularly useful for branches near the boiler where the drop in the main has not yet been produced.

Orifice systems using a vacuum pump operate successfully with the ordinary low vacuum type of pump producing 8 to 10 in. of vacuum. They are controlled by various means to regulate the steam pressure. One method is by a thermostat located on the roof to govern the steam pressure by a combination of outside and inside temperatures; another, useful on systems without traps and vacuum pumps, controls the steam pressure manually from temperature indication stations in the building, or automatically by a thermostatically-controlled pressure reduction valve or draft regulator on the boiler; with oil or gas firing, the on-and-off control or a boiler pressure control may be used.

ZONE CONTROL

Certain portions of a building may require more heat at times than others but if the whole building is on one general control, such as would occur with a single piping system with an on-and-off control or with the sub-atmospheric or the orifice systems, it would be necessary to supply sufficient heat to accommodate the coldest portion of the building even though some sections would be overheated. By zoning, each section of a building may be controlled separately.

The sides of the building with different exposures should be considered first, because of the varying effects of the wind and sun. With the prevailing winter winds from the northwest, a simple zoning would place the north and west sides of the building on one system and the south and east sides on another. If the building is large enough to justify the expenditure, a better arrangement would be to place all north walls on one zone, all west walls on a second, all east walls on a third, and all south walls on a fourth.

In case of high buildings, the lower 8 or 10 stories may be well protected from wind by surrounding buildings, the next 10 stories may have moderate exposure, and above this there may be an unobstructed exposure to gales. On still days the heat demands vertically will vary little, but on windy days there will be a marked difference in the heat requirements for the different horizontal sections. In addition, the *chimney effect* caused by the difference in density between the warm air on the inside of a building and the colder air on the outside will give an air movement which will require zoning to correct. Where such conditions are encountered, the building should be divided horizontally as well as vertically. An arrangement of this character would give 12 zones, namely: north, east, south, and west lower zones; similar middle zones; and similar top zones. Each zone should constitute an individual and separate system of piping with its own supply steam valve (controlled by thermostats in its respective zone) and with its own return or vacuum pump, if one is used. Certain interior areas, such as basements, light well walls and other locations where sun and wind do not affect the conditions, should be placed in still another zone if the most economical results are to be secured.

Zoning has advantages even where individual thermostatic radiator

control is installed whether this be of pneumatic, electric, or the self-contained radiator valve type. By operating the different zones to parallel outside temperature requirements, a large part of the load is taken off the thermostatic controls, they make fewer operations and the radiator follows a more even temperature instead of fluctuating from extreme hot to extreme cold.

CONDENSATION RETURN PUMPS

Condensation return pumps are generally required when the elevation of the boiler with respect to the heating units is such that the condensate will not return by gravity, or when the boiler pressure is greater than that

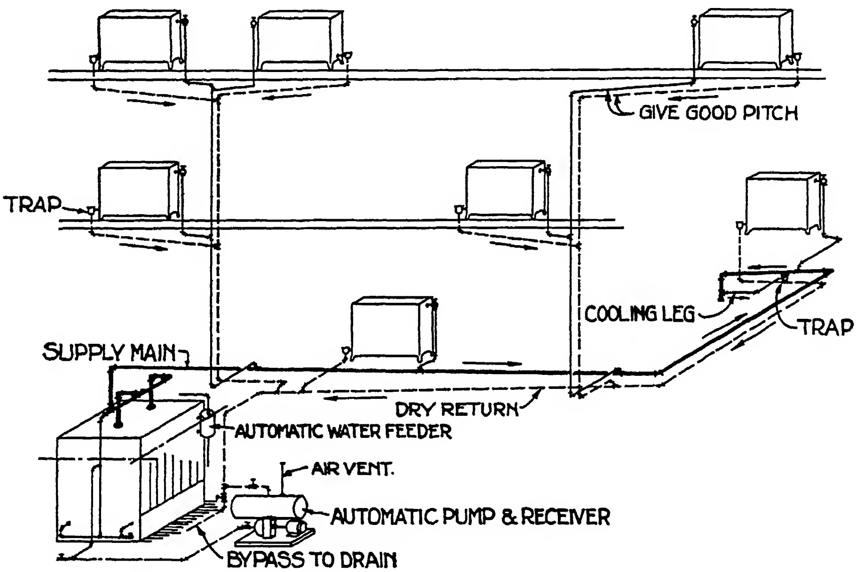


FIG. 19. TYPICAL INSTALLATION USING CONDENSATION PUMP

supplied the heating units, as in a high-pressure boiler installation supplying steam through a reducing valve to the heating units. The condensate is commonly returned by gravity to a receiver, vented to the atmosphere, from which it flows to the pump.

Condensation return pumps are assembled with tank or receiver and arranged for either continuous operation or for automatic starting and stopping by float control. Any style of water pump may be employed for this service, the power available determining whether the mode of drive shall be steam or electric. The motor-driven, automatic, centrifugal pump and receiver has found wide acceptance in practice for low pressure heating systems.

Fig. 19 shows a typical installation using an automatic condensation return pump and vented receiver. A float control operates the pump whenever sufficient water accumulates. Condensation return pumps are

suitable for use on systems in which the returns are under atmospheric pressure. These include atmospheric systems, orifice systems with open returns, and certain types of vapor systems which operate within a few ounces of atmospheric pressure, but ordinarily do not carry any sub-atmospheric pressure. They may also be used on one-pipe and two-pipe gravity steam systems with a proper arrangement for venting the receiver. In discharging to waste, there is no object in using a condensation pump unless the discharge must be elevated.

VACUUM PUMPS

A vacuum heating pump is employed to create a vacuum on the return end of a system to remove air and water and to return the condensate to the boiler or to some other intercepting device that may be employed in plants having mixed systems of heating and other services. Pumps of this classification may be driven by steam or electricity; they may be continuous in operation, or automatic with float or vacuum control in one or more combinations.

Return line vacuum pumps are classified as follows:

a. Those which perform the function of air separation under atmospheric pressure.

b. Those which perform the function of air separation under a partial vacuum.

Pumps coming under the first classification will handle vacuum steam system condensation coming back by gravity at any temperature up to 205 F without either the sealing or the hurling water flashing into steam. These pumps, to operate under a combined water level and vacuum control, must be equipped with a float-control receiver between the vacuum pump and the system, but where they are intended for continuous operation, they do not require a receiver. Such pumps employ a single vacuum producer which removes the condensate and air from the system and delivers it into a separating chamber under atmospheric pressure from which the condensate is delivered to the boiler or feed water heater. They are constructed on one of the following evacuating and discharge principles:

1. Hydraulic vacuum producer with one pump impeller.
2. Hydraulic vacuum producer with two pump impellers.
3. Water displacement vacuum producer with two pump impellers.
4. Piston displacement vacuum producer with one pump piston.

The second classification of pumps will handle vacuum steam system condensation coming back by gravity at any temperature not exceeding 190 F without the flashing into steam of either the sealing or the hurling water. In order to operate under a combined water-level and vacuum control, these pumps must be equipped with a float-control receiver between the vacuum pump and the system; where intended for continuous operation they do not require a receiver. Such pumps employ a vacuum producing impeller which removes air from the receiver or heating system under a partial vacuum and delivers it through an air separator against atmospheric pressure. The condensate is removed from the receiver under a partial vacuum by a separate impeller and is delivered to the boiler or feed water heater. For evacuating and dis-

charge, a water displacement vacuum producer with two pump impellers is used.

Receiver Capacities for Vacuum Pumps

Where receivers are used in connection with vacuum pumps there is a definite relation between the capacity of the receiver and the capacity of the pump. The receiver should have a capacity of not less than 1½ times the volumetric quantity of condensation per minute and should not have such a capacity that the pump will empty the receiver in less than half a minute. Receivers of larger capacities will result in less frequent periods of operation.

Duplex Vacuum Pumps

Duplex vacuum pumps consist of two pumps mounted on a common or separate bedplate and having a common receiver. Duplex pumps now on the market include a motor-drive on one pump and a low-pressure steam-turbine-drive on the other, the exhaust from the turbine being used in the heating system as long as the system has the capacity to condense all of it. When the system capacity has fallen to a point where it no longer can completely condense the turbine exhaust, the equipment automatically goes over onto electric operation but as soon as the steam demand in the system rises again the turbine is thrown into operation and the motor is cut out.

Piston Displacement Vacuum Pumps

Piston displacement return-line vacuum heating pumps may be either power or steam driven. They should be provided with mechanical lubricators and their piston speed in feet per minute should not exceed 20 times the square root of the number of inches in their stroke. While the volumetric displacement for such pumps was formerly figured at 8 to 10 times the volumetric flow of condensation to be handled, the more efficient thermostatic traps used today in connection with vacuum heating systems make it possible to change this proportion so that the volumetric displacement of these pumps may not be less than 6 times the volume of condensation.

Vacuum Pump Controls

In the ordinary vacuum system the vacuum pump is controlled by a vacuum regulator which cuts in when the vacuum drops to the lowest point desired and which cuts out when the vacuum has been increased to the highest point. This is done largely to eliminate the constant starting and stopping of the vacuum pump which would occur if the vacuum were maintained constant. In addition to this control, a float control is included which will automatically start the pump whenever sufficient condensation accumulates in the receiver, regardless of the vacuum in the system. This arrangement makes the vacuum pump primarily a condensation pump and secondarily an air pump.

On the sub-atmospheric systems the high vacuum pump is controlled by a differential regulator which keeps the vacuum in the return line always a few inches higher than that in the steam line and in the radiators.

TRAPS

Traps are used for draining the condensate from radiators, steam piping systems, kitchen equipment, laundry equipment, hospital equipment, drying equipment and many other kinds of apparatus. The usual functions of a trap are to allow the passage of condensate and to prevent the passage of steam. In addition to these functions, traps are frequently required to allow the passage of air as well as condensate. Traps are also required to allow the passage of air and to prevent the passage of either water or steam, or both.

In addition, traps are used for returning condensate either by gravity, by steam pressure, or by both, to a boiler or other point of disposal, and for lifting condensate from a lower to a higher elevation, or for handling condensate from a lower to a higher pressure.

The fundamental principle upon which the operation of practically all traps depends is that the pressure within the trap at the time of discharge shall be equal to, or slightly in excess of, the pressure against which the trap must discharge, including the friction head, velocity head and static head on the discharge side of the trap. If the static head is in favor of the trap discharge it is a minus quantity and may be deducted from the other factors of the discharge head.

Traps may be classified according to the principle of operation as (1) float, (2) bucket, (3) thermostatic, or (4) tilting traps.

Float Traps. A discharge valve is operated by the rise and fall of a float due to the change of water level in the trap. When the trap is empty the float is in its lowest position, and the discharge valve is closed. A gage glass indicates the height of water in the chamber.

Unless float traps are well made and proportioned there is danger of considerable steam leakage through the discharge valve due to unequal expansion of the valve and seat and the sticking of moving parts. The discharge from a float trap is usually continuous since the height of the float, and consequently the area of the outlet, is proportional to the amount of water present.

Bucket Traps. Bucket traps are of two types, the upright and inverted, and although they are both of the open float construction, their operating principle is entirely different. In the *upright bucket* trap, the water of condensation enters the trap and fills the space between the bucket and the walls of the trap. This causes the bucket to float and forces the valve against its seat, the valve and its stem usually being fastened to the bucket. When the water rises above the edges of the bucket it flows into it and causes it to sink, thereby withdrawing the valve from its seat. This permits the steam pressure acting on the surface of the water in the bucket to force the water to a discharge opening. When the bucket is emptied it rises and closes the valve and another cycle begins. The discharge from this type of trap is intermittent.

In the *inverted bucket* trap, steam floats the inverted submerged bucket and closes the valve. Water entering the trap fills the bucket which sinks and through compound leverage opens the valve, and the trap discharges. It is impossible to install a water gage glass on an inverted bucket trap, but if visual inspection is necessary, a gage glass can be placed on the line leading to the trap. No air relief cocks can be used, but this is unnecessary, as the elimination of air is automatically taken care of by air passing through the vent in the top of the inverted bucket regardless of temperature.

Thermostatic Traps. Thermostatic traps are of two types, those in which the discharge valve is operated by the relative expansion of metals, and those in which the action of a volatile liquid is utilized for this purpose. Thermostatic traps of large capacity for draining blast coils or very large radiators, are called *blast* traps.

Tilting Traps. With this type of trap, water enters a bowl and rises until its weight over-balances that of a counter-weight, and the bowl sinks to the bottom. As the bowl sinks, a valve is opened thus admitting live steam pressure on the surface of the water

and the trap then discharges. After the water is discharged, the counter-weight sinks and raises the bowl, which in turn closes the valve and the cycle begins again. Tilting traps are necessarily intermittent in operation. They are not ordinarily equipped with glass water gages, as the action of the trap shows when it is filling or emptying. The air relief of tilting traps is taken care of by the valves of the trap.

Thermostatic traps are generally used for draining radiators and heaters, except for very large capacities where bucket, float or blast-type thermostatic traps are used. Thermostatic traps for this service usually pass both condensate and air and in the case of float and upright bucket traps the air is usually relieved through an auxiliary thermostatic trap in a by-pass around the main trap. Sometimes this auxiliary air trap is an integral part of the trap.

Blast-type thermostatic traps are sometimes used on vacuum heating

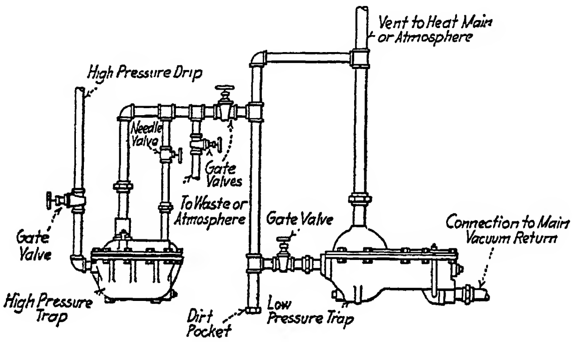


Fig. 20. Method of Discharging High-Pressure Apparatus into Low-Pressure Heating Mains and Vacuum Return Mains through a Low-Pressure Trap

systems for connecting old one- or two-pipe gravity systems in parallel with vacuum return line systems, in which case the blast-type thermostatic traps should not be provided with auxiliary air by-pass, as the action of this will allow the vacuum to draw air into the old system through its air valves, especially when the steam is wholly or partially cut off. The air from the returns of such old systems should be relieved just ahead of the traps by means of quick-venting automatic air valves, preferably of the non-return type, especially if the other air valves on the old system are non-return valves.

Tilting traps used for discharging to a higher or a lower pressure are provided with two or three valves operated by the action of the trap. In the case of the two-valve tilting traps, one valve closes a steam inlet and the other valve opens a vent outlet while the trap is filling, and as soon as the trap dumps the first valve opens the steam inlet and the second valve closes the vent outlet, while the trap discharges. In this type of trap there must be a swinging check-valve on each side of the trap, in addition to the usual by-pass, to prevent the pressure in the trap, while discharging, from backing up through the inlet and the pressure in the discharge line from backing up into the trap while it is filling. This type of trap will blow steam out through the vent while filling, if the

pressure on the inlet side is sufficient, and should not be used, therefore, with such pressures unless the vent is properly piped back into the return to a feed water heater, a condenser or a perforated pipe in the bottom of the receiver to which the trap discharges in such a way as to prevent the escape of the steam that comes in with the condensate and passes

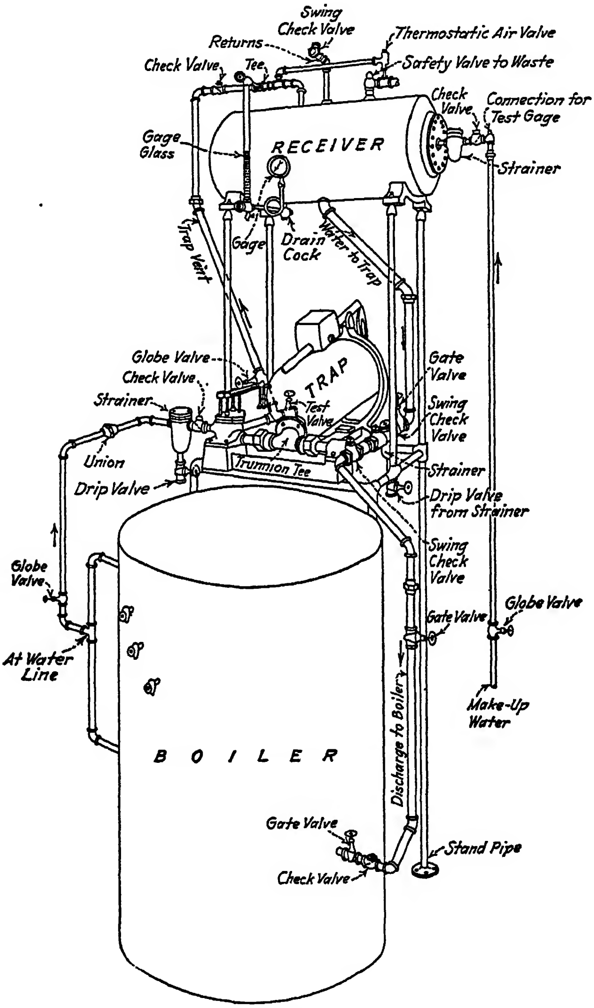


FIG. 21. RETURN TRAP AND RECEIVER FOR AUTOMATIC BOILER FEED

through the vent. In the three-valve traps of this type there is an extra valve for closing the discharge while the trap is filling.

High pressure traps should not discharge directly into a vacuum return because of the vapor formed by the re-evaporation of a part of the hot condensation. Fig. 20 shows a method which may be used for disposing of the greater part of the vapor of re-evaporation.

Automatic Return Traps

In the general heating plant, where thermostatic traps are installed on the heating units, it becomes necessary to provide a means for returning the water of condensation to the boiler, if a condensation or vacuum pump is not used. When the return main can be kept sufficiently high above the boiler water line for all operating conditions, the water of condensation will flow back by gravity, and no mechanical device is required. But actually this does not work out in practice. It follows, therefore, that a direct return trap is needed for the handling of the condensation even though it may not be called into action except under some operating condition where the pressure differential exceeds the static head provided. The installation of a direct return trap assures safety for such systems, and the operation of the plant under varying conditions.

Automatic return traps, sometimes called alternating receivers, may be of the counterbalanced, tilting type, or spring actuated. These consist of a small receiver with an internal float, and when the condensate will not flow into the boiler under pressure, it will feed into the receiver of the trap, and in so doing, raise or tilt the float or mechanism which actuates a steam valve automatically. This admits steam to the receiver, at boiler pressure, and the equalizing of the pressures which follows allows the water to flow into the boiler. Fig. 21 shows a direct return tilting trap and receiver properly connected for automatically feeding a boiler from a system of returns delivering the condensate to the receiver.

Chapter 32

STEAM SYSTEM PIPING

Flow of Steam in Pipes, Pipe Sizes, Initial Pressure, Pressure Drop, Maximum Velocity, Reaming, Equivalent Length of Run, Tables for Pipe Sizing, Sizing One-Pipe Gravity Air Vent Systems, Two-Pipe Gravity Air Vent Systems, Two-Pipe Vapor Systems, Atmospheric Systems, Vacuum Systems, Sub-Atmospheric Systems, Orifice Systems, Pressure-Reducing Valves, Expansion in Steam and Return Lines, Piping Connections and Details, Boiler Connections, Hartford Return Connection

THE design of a steam heating system may be divided into four parts, namely, (1) the details of the heating units, (2) the arrangement of the general piping scheme, (3) the details of connections, and (4) the sizing of the lines. Items 1 and 2 are covered in Chapters 30 and 31, respectively, while this chapter considers the two latter items.

The functions of piping are to supply the heating units with steam and to remove the condensation. In some systems both the air and condensation are removed from the heating units by the return piping. To accomplish this effectively, the distribution of the steam should be efficient and equitable, without noise, and the returns should be as short as possible. When air is handled its escape should be facilitated to the utmost since an air-bound system will not heat properly. Condensation takes place in a steam system not only in the heating units, but throughout the piping system as well, and the returns also condense any steam or vapor that may be contained. At the same time part of the condensation may flash back into steam when the vacuum or pressure in the return is considerably below the steam pressure.

It is essential that steam piping systems not only distribute steam at full load but also at partial loads, as the average winter demand is less than half of the demand in most severe outside temperatures. Furthermore, in heating up rapidly the load on the steam main may exceed the maximum operating load even in extreme weather, due to the necessity of raising the temperature of the metal in the system to the steam temperature. This may require more heat than would be emitted from the system itself after it once is thoroughly heated.

STEAM FLOW IN PIPES

The rate of flow of dry steam or steam with a small amount of water flowing in the same direction is in accordance with the general laws of gas flow and is a function of the length and diameter of the pipe, the density of the steam and the pressure drop through the pipe. This relationship has been established by Babcock in the following formula:

$$P = 0.0000000367\left(1 + \frac{3.6}{d}\right)\frac{W^2L}{Dd^5} \tag{1}$$

or

$$W = 5220\sqrt{\frac{PDd^5}{\left(1 + \frac{3.6}{d}\right)L}} \tag{2}$$

where

P = loss in pressure, pounds per square inch.
d = inside diameter of pipe, inches.
L = length of pipe, feet.
D = weight of 1 cu ft of steam.
W = weight of steam flowing per hour, pounds.

Example 1. How much steam will flow per hour through 100 ft of 2-in. pipe if the initial pressure is 1.3 lb per square inch and the pressure drop is 1 oz?

Solution. $P = \frac{1}{16} = 0.0625$ lb; $d = 2.067$ in. (Table 1, Chapter 34); $L = 100$ ft; $D = 0.04038$ lb (Table 5, Chapter 41). Substituting these values in Formula 2:

$$W = 5220\sqrt{\frac{0.0625 \times 0.04038 \times 2.067^5}{\left(1 + \frac{3.6}{2.067}\right)100}} = 97.2 \text{ lb per hour.}$$

Formula 2 does not allow for entrained water in low-pressure steam, condensation in pipe, and roughness in commercial pipe as found in practice.

The latent heat of steam (h_{fg}) at atmospheric pressure (Table 5, Chapter 41) is 970.2 Btu per pound. Inasmuch as the heat emission of an equivalent square foot of heating surface (radiation) is 240 Btu, 1 lb of steam at this pressure will supply $\frac{970.2}{240}$ or 4.04 sq ft of equivalent heating surface. This figure is usually taken as 4 even. In Example 1, the weight of steam flowing per hour would therefore supply 4 × 97.2 or 388.8 sq ft of equivalent heating surface.

PIPE SIZES

The determination of pipe sizes for steam heating depends on the following principal factors:

1. The initial pressure and the total pressure drop which may be allowed between the source of supply and the end of the return system.
2. The maximum velocity of steam allowable for quiet and dependable operation of the system.
3. The equivalent length of the run from the boiler or source of steam supply to the farthest heating unit.
4. Unusual conditions in the building to be heated.

Initial Pressure and Pressure Drop

Theoretically there are several factors to be considered, such as initial pressure and pressure required at the end of the line, but it is most important that (1) the total pressure drop does not exceed the initial pressure of the system; (2) the pressure drop is not so great as to cause excessive velocities; (3) there is a constant initial pressure, except on systems

TABLE 1. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR ONE-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

PIPE SIZE INCHES	VELOCITY FEET PER SECOND	PRESSURE DROP OUNCES PER 100 FT	CAPACITY		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
1	14.1	0.68	45	10,961	11.3
1¼	17.6	0.66	98	23,765	24.5
1½	20.0	0.66	152	36,860	38.0
2	23.0	0.57	288	69,840	72.0
2½	26.0	0.54	464	112,520	116.0
3	29.0	0.48	799	193,600	199.8
3½	31.0	0.44	1144	277,000	286.0
4	32.0	0.39	1520	368,000	380.0

INSTRUCTIONS FOR USING TABLE 1

1. Capacities given in Table 1 should never be exceeded on one-pipe risers.
2. Capacities are based on ¼ lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5).

specially designed for varying initial pressures, such as the sub-atmospheric, the orifice, and the vapor systems which normally operate under partial vacuums; (4) there is sufficient difference in level, for gravity return systems, between the lowest point on the steam main, the heating units, and the dry return, when considered in relation to the boiler water line.

All systems should be designed for a low initial pressure and a reasonably small pressure drop for two reasons: *first*, the present tendency in steam heating unmistakably points toward a constant lowering of pressures even to those below atmospheric; *second*, a system designed in this manner will operate under higher pressures without difficulty. When a system designed for a relatively high initial pressure and a relatively high pressure drop is operated at a lower pressure, it is likely to be noisy and have poor circulation.

The total pressure drop should never exceed one-half of the initial pressure when condensate is flowing in the same direction as the steam. Where the condensate must flow counter to the steam, the governing factor is the velocity permissible without interfering with the condensate flow. Laboratory experiments limit this to the capacities given in Tables 1 and 2 for vertical risers and in Table 3 for horizontal pipes at varying grades.

Maximum Velocity and Reaming

The capacity of a steam pipe in any part of a steam system depends upon the quantity of condensation present, the direction in which the condensate is flowing, and the pressure drop in the pipe. Where the

TABLE 2. MAXIMUM ALLOWABLE CAPACITIES OF UP-FEED RISERS FOR TWO-PIPE LOW PRESSURE STEAM

Based on A. S. H. V. E. Research Laboratory Tests

Pipe Size Inches	Velocity Feet per Second	Pressure Drop Ounces per 100 Ft	Capacity		
			Sq Ft Radiation	Btu per Hour	Lb Steam per Hour
A	*B*	*C*	*D*	*E*	*F*
¾	20	----	40	9550	10.0
1	23	1.78	74	17,900	18.45
1¼	27	1.57	151	36,500	37.65
1½	30	1.48	228	55,200	57.0
2	35	1.33	438	106,100	109.5
2½	38	1.16	678	164,100	169.4
3	41	0.95	1129	273,500	282.2
3½	42	0.81	1548	375,500	387.0
4	43	0.71	2042	495,000	510.5

INSTRUCTIONS FOR USING TABLE 2

1. The capacities given in this table should never be exceeded on two-pipe risers.
2. Capacities are based on ¼ lb condensation per square foot equivalent radiation and actual diameter of standard pipe.
3. All pipe should be well reamed and free from constrictions. Fittings should be up to size. (See Tables 4 and 5).

quantity of condensate is limited and is flowing in the same direction as the steam, only the pressure drop need be considered. When the condensate must flow against the steam, even in limited quantity, the velocity of the steam must not exceed limits above which the disturbance between the steam and the counter-flowing water may produce objectionable sounds, such as water hammer, or may result in the retention of water in certain parts of the system until the steam flow is reduced sufficiently to permit the water to pass. The velocity at which such disturbances take place is a function of (1) the pipe size, whether the pipe runs horizontally or vertically, (2) the pitch of the pipe if it is run horizontally, and (3) the quantity of condensate flowing against the steam.

Two factors of uncertainty always exist in determining the capacity of any steam pipe. The first is variation in manufacture, which apparently cannot be avoided and which caused an actual difference of 20 per cent in the capacity of a 1 in. pipe in experiments carried on at the A.S.H.V.E. Research Laboratory (Table 4). The second is the reaming of the ends of the pipe after cutting, which, experiments indicate, might reduce the capacity of a 1 in. pipe as much as 28.7 per cent (Table 5). All of the capacity tables given in this chapter include a factor of safety. However, the pipe on which Table 4 is based showed no particular defects or constrictions on the inside, and the factor of safety referred to does not cover abnormal defects or constrictions *nor does it cover pipe not properly reamed.*

TABLE 3. COMPARATIVE CAPACITY OF STEAM LINES AT VARIOUS PITCHES[a]

Pitch of Pipe in Inches per 10 Ft

PITCH OF PIPE	¼ IN.		½ IN.		1 IN.		1½ IN.		2 IN.		3 IN.		4 IN.		5 IN.	
Pipe Size Inches	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.	Sq Ft Rad. Based on 240 Btu	Max. Vel.
¾	25.0	12	30.3	14	37.3	18	40.4	19	42.5	20	46.1	21	47.5	22	49.3	23
1	45.8	12	52.6	15	63.0	17	70.0	20	75.2	22	83.0	23	87.9	25	90.2	26
1¼	104.9	18	117.2	20	133.0	23	144.5	25	154.0	27	165.0	28	172.6	29	178.2	31
1½	142.6	18	159.0	21	181.0	23	196.5	25	209.3	27	224.0	28	234.8	30	242.6	31
2	236.0	19	263.5	20	299.5	23	325.5	25	346.5	27	371.5	28	388.4	29	401.1	30

[a]Data from A.S.H.V.E. Research Laboratory.

Equivalent Length of Run

All tables for the flow of steam in pipes, based on pressure drop, must allow for the friction offered by the pipe as well as for the additional resistance of the fittings and valves. These resistances generally are stated in terms of straight pipe; in other words, a certain fitting will produce a drop in pressure equivalent to so many feet of straight run of the same size of pipe. Table 6 gives the number of feet of straight pipe usually allowed for the more common types of fittings and valves. In all pipe sizing tables in this chapter the *length of run* refers to the *equivalent length of run* as distinguished from the *actual length* of pipe in feet. The length of run is not usually known at the outset; hence it is necessary to assume some pipe size at the start. Such an assumption frequently is considerably in error and a more common and practical method is to assume the length of run and to check this assumption after the pipes are sized. For this purpose the length of run usually is taken as double the actual length of pipe.

TABLE 4. PER CENT DIFFERENCE IN CAPACITY DUE TO VARIATION OF PIPE SIZE AND SMOOTHNESS[a]

	MAXIMUM CONDENSATION, LB PER HOUR			
Size of Pipe	¾″	1″	1¼″	1½″
Minimum	14.00	24.89	45.42	70.50
Maximum	15.20	30.08	52.08	82.00
Per Cent Variation	8.6	20.8	14.7	16.3

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEER'S Research Laboratory.

TABLE 5. EFFECT OF REAMING ENTRANCE TO ONE-INCH ONE-PIPE RISERS[a]

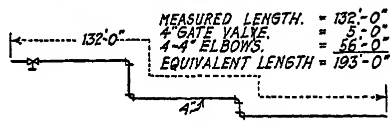
	MAXIMUM CAPACITY OF RISER	PER CENT DECREASE
Reamed entrances	24.7 lb per hour	0.0
Rounded entrances	23.9 lb per hour	3.2
Squared entrances	22.2 lb per hour	10.1
Three wheel cutter	19.2 lb per hour	22.2
Single wheel cutter	17.6 lb per hour	28.7

[a]Data from AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS Research Laboratory.

TABLE 6. LENGTH IN FEET OF PIPE TO BE ADDED TO ACTUAL LENGTH OF RUN—OWING TO FITTINGS—TO OBTAIN EQUIVALENT LENGTH

SIZE OF PIPE INCHES	ST'D. ELBOW	SIDE OUTLET TEE	GATE VALVE	GLOBE VALVE	ANGLE VALVE
	Length in Feet to be Added in Run				
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
3½	12	31	4	39	19
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
7	26	55	10	82	37
8	31	63	12	94	42
9	35	69	13	105	47
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

Example of length in feet of pipe to be added to actual length of run.



TABLES FOR PIPE SIZING[1]

Factors determining the size of a steam pipe and its allowable limit of capacity are as follows:

1. Pipe condensate flowing with steam.
2. Pipe condensate flowing against steam.
3. Pipe and radiator condensate flowing with steam.
4. Pipe and radiator condensate flowing against steam.

It is apparent that (3) and (4) are practically limited to one-pipe systems while (1) and (2) cover all other systems.

Tables 7 and 8, worked out for determining pipe sizes, have their columns lettered continuously, Columns *A* through *L* being in Table 7, and *M* through *EE* in Table 8. In the following text, reference made to columns will be by letter. The tables are based on the actual inside diameters of the pipe and the condensation of ¼ lb of steam per square foot of equivalent direct radiation[2] (*abbreviated EDR*) per hour. The drops indicated are drops in pressure per 100 ft of equivalent length of run. The pipe is assumed to be well reamed without unusual or noticeable defects.

[1]Pipe size tables in this chapter have been compiled in simplified and condensed form for the convenience of the user; at the same time all of the information contained in previous editions of THE GUIDE has been retained. Values of pressure drops, formerly expressed in ounces, are now expressed in fractions of a pound.

[2]As steam system design has materially changed in recent years so that 240 Btu no longer expresses the heat of condensation from a square foot of radiator surface per hour, and as present day heating units have different characteristics from older forms of radiation, it is the purpose of THE GUIDE to gradually eliminate the empirical expression *square foot of equivalent direct radiation*, *EDR*, and to substitute a logical unit based on the Btu. The Committee on Nomenclature is recommending new terms to express the equivalent of 1000 Btu, and 1000 Btu per hour, for approval by the A.S.H.V.E. In this edition of THE GUIDE, the *equivalent square foot* has been retained in the tables for the sizing of steam heating systems.

TABLE 7. STEAM PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Radiation

(Reference to this table will be by column letter *A* through *L*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

	CAPACITIES OF STEAM MAINS AND RISERS								SPECIAL CAPACITIES FOR ONE-PIPE SYSTEMS ONLY		
	DIRECTION OF CONDENSATION FLOW IN PIPE LINE										
PIPE SIZE IN.	With the Steam in One-Pipe and Two-Pipe						Against the Steam Two-Pipe Only		Supply Risers Up-Feed	Radiator Valves and Vertical Connections	Radiator and Riser Run-outs
	1/32 lb Drop	1/24 lb Drop	1/16 lb Drop	1/8 lb Drop	1/4 lb Drop	1/2 lb Drop	Vertical	Horizontal			
A	*B*	*C*	*D*	*E*	*F*	*G*	*H*[a]	*I*[c]	*J*[b]	*K*	*L*[c]
3/4	....	....	30	....	....	....	30	....	25	....	....
1	39	46	56	79	111	157	56	26	45	20	20
1¼	87	100	122	173	245	346	122	58	98	55	55
1½	134	155	190	269	380	538	190	95	152	81	81
2	273	315	386	546	771	1,091	386	195	288	165	165
2½	449	518	635	898	1,270	1,797	635	395	464	....	260
3	822	948	1,163	1,645	2,326	3,289	1,129	700	799	....	475
3½	1,228	1,419	1,737	2,457	3,474	4,913	1,548	1,150	1,144	....	745
4	1,738	2,011	2,457	3,475	4,914	6,950	2,042	1,700	1,520	....	1,110
5	3,214	3,712	4,546	6,929	9,092	12,858	....	3,150	....	....	2,180
6	5,276	6,094	7,642	10,553	14,924	21,105	....	....	....	....	....
8	10,983	12,682	15,533	21,967	31,066	43,934	....	....	....	....	....
10	20,043	23,144	28,345	40,085	56,689	80,171	....	....	....	....	....
12	32,168	37,145	45,492	64,336	90,985	128,672	....	....	....	....	....
16	60,506	69,671	84,849	121,012	169,879	240,245	....	....	....	....	....
	All Horizontal Mains and Down-Feed Risers						Up-Feed Risers	Mains and Undripped Run-outs	Up-Feed Risers	Radiator Connections	Run-outs Not Dripped

Note.—All drops shown are in pounds per 100 ft of equivalent run—based on pipe properly reamed.

[a]Do not use Column *H* for drops of 1/24 or 1/32 lb; substitute Column *C* or Column *B* as required.

[b]Do not use Column *J* for drop of 1/32 lb except on sizes 3 in and over; below 3 in. substitute Column *B*.

[c]On radiator runouts over 8 ft long increase one pipe size over that shown in Table 7.

Table 7 may be used for sizing piping for steam heating systems by determining the allowable or desired pressure drop per 100 equivalent feet of run and reading from the column for that particular pressure drop. This applies to all steam mains on both one-pipe and two-pipe systems, vapor systems, and vacuum systems. Columns *B* to *G* inclusive are used where the steam and condensation flow in the same direction, while Columns *H* and *I* are for cases where the steam and condensation flow in opposite directions, as in risers and runouts that are not dripped. Columns *J*, *K*, and *L* are for one-pipe systems and cover riser, radiator valve and vertical connection sizes, and radiator and runout sizes, all of which are based on the critical velocities of the steam to permit the counter flow of condensation without noise.

Sizing of return piping may be done with the aid of Table 8 where pipe capacities for wet, dry, and vacuum return lines are shown for the pressure drops per 100 ft corresponding to the drops in Table 7. *It is customary to use the same pressure drop on both the steam and return sides of a system.*

TABLE 8. RETURN PIPE CAPACITIES

Capacity Expressed in Square Feet of Equivalent Radiation

(Reference to this table will be by column letter *M* through *EE*)

This table is based on pipe size data developed through the research investigations of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

CAPACITY OF RETURN MAINS AND RISERS

Pipe Size Inches	1/32 lb Drop per 100 Ft			1/24 lb Drop per 100 Ft			1/16 lb Drop per 100 Ft			1/8 lb Drop per 100 Ft			1/4 lb Drop per 100 Ft			1/2 lb Drop per 100 Ft		
	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.	Wet	Dry	Vac.
M	*N*	*O*	*P*	*Q*	*R*	*S*	*T*	*U*	*V*	*W*	*X*	*Y*	*Z*	*AA*	*BB*	*CC*	*DD*	*EE*
	Mains																	
3/4	....	....	....	....	....	326	....	....	400	....	....	568	....	....	800	....	....	1,130
1	500	248	....	580	285	570	700	320	700	1,000	412	994	1,400	460	1,400	....	....	1,977
1 1/4	850	520	....	990	595	976	1,200	670	1,200	1,700	868	1,704	2,400	962	2,400	....	....	3,390
1 1/2	1,350	822	....	1,570	943	1,547	1,900	1,058	1,900	2,700	1,362	2,696	3,800	1,512	3,800	....	....	5,370
2	2,800	1,880	....	3,240	2,140	3,256	4,000	2,300	4,000	5,600	2,960	5,680	8,000	3,300	8,000	....	....	11,300
2 1/2	4,700	3,040	....	5,300	3,470	5,453	6,700	3,800	6,700	9,400	4,900	9,510	13,400	5,450	13,400	....	....	18,925
3	7,500	5,840	....	8,500	6,250	8,710	10,700	7,000	10,700	15,000	9,000	15,190	21,400	10,000	21,400	....	....	30,230
3 1/2	11,000	7,880	....	13,200	8,800	13,020	16,000	10,000	16,000	22,000	12,900	22,710	32,000	14,300	32,000	....	....	45,200
4	15,500	11,700	....	18,300	13,400	17,910	22,000	15,000	22,000	31,000	19,300	31,220	44,000	21,500	44,000	....	....	62,180
5	....	....	....	....	....	31,500	....	....	38,700	....	....	54,920	....	....	77,400	....	....	109,300
6	....	....	....	....	....	50,450	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,100
	Risers																	
3/4	....	190	....	....	190	570	....	190	700	....	190	994	....	190	1,400	....	....	1,977
1	....	450	....	....	450	976	....	450	1,200	....	450	1,704	....	450	2,400	....	....	3,390
1 1/4	....	990	....	....	990	1,547	....	990	1,900	....	990	2,696	....	990	3,800	....	....	5,370
1 1/2	....	1,500	....	....	1,500	3,256	....	1,500	4,000	....	1,500	5,680	....	1,500	8,000	....	....	11,300
2	....	3,000	....	....	3,000	5,453	....	3,000	6,700	....	3,000	9,510	....	3,000	13,400	....	....	18,925
2 1/2	....	....	....	....	....	8,710	....	....	10,700	....	....	15,910	....	....	21,400	....	....	30,230
3	....	....	....	....	....	13,020	....	....	16,000	....	....	22,710	....	....	32,000	....	....	45,200
3 1/2	....	....	....	....	....	17,910	....	....	22,000	....	....	31,220	....	....	44,000	....	....	62,180
4	....	....	....	....	....	31,500	....	....	38,700	....	....	54,920	....	....	77,400	....	....	109,300
5	....	....	....	....	....	50,450	....	....	62,000	....	....	88,000	....	....	124,000	....	....	175,100

Example 2. What pressure drop should be used for the steam piping of a system if the measured length of the longest run is 500 ft and the initial pressure is not to be over 2 lb gage?

Solution. It will be assumed, if the measured lengtn of the longest run is 500 ft, that when the allowance for fittings is added the equivalent length of run will not exceed 1,000 ft. Then, with the pressure drop not over one-half of the initial pressure, the drop could be 1 lb or less. With a pressure drop of 1.lb and a length of run of 1,000 ft, the drop per 100 ft would be 1⁄10 lb, while if the total drop were ½ lb, the drop per 100 ft would be 1⁄20 lb. In the first instance the pipe could be sized according to Column *D* for 1⁄16 lb per 100 ft, and in the second case, the pipe could be sized according to Column *C* for 1⁄24 lb. On completion of the sizing, the drop could be checked by taking the longest line and actually calculating the equivalent length of run from the pipe sizes determined. If the calculated drop is less than that assumed, the pipe size is all right; if it is more, it is probable that there are an unusual number of fittings involved, and either the lines must be straightened or the column for the next lower drop must be used and the lines resized. Ordinarily resizing will be unnecessary.

SIZING ONE-PIPE GRAVITY AIR VENT SYSTEMS

One-pipe gravity air vent systems in which the equivalent length of run does not exceed 200 ft should be sized as follows:

1. *For the steam main and dripped runouts to risers* where the steam and condensate flow in the same direction, use 1⁄16 lb drop (Column *D*).

2. *Where the riser runouts are not dripped* and the steam and condensation flow in opposite directions, and *also in the radiator runouts* where the same condition occurs, use Column *L*.

3. *For up-feed steam risers* carrying condensation back from the radiators, use Column *J*.

4. *For down-feed systems the main risers* of which do not carry any radiator condensation, use Column *H*.

5. *For the radiator valve size and the stub connection,* use Column *K*.

6. For the *dry return main,* use Column *U*.

7. For the *wet return main* use Column *T*.

On systems exceeding an equivalent length of 200 ft, it is suggested that the total drop be not over ¼ lb. The return piping sizes should correspond with the drop used on the steam side of the system. Thus, where 1⁄24 lb drop is being used, the steam main and dripped runouts would be sized from Column *C;* radiator runouts and undripped riser runouts from Column *L;* up-feed risers from Column *J*; the main riser on a down-feed system from Column *C* (it will be noted that if Column *H* is used the drop would exceed the limit of 1⁄24 lb); the dry return from Column *R;* and the wet return from Column *Q*.

With a 1⁄32-lb drop the sizing would be the same as for 1⁄24 lb except that the steam main and dripped runouts would be sized from Column *B*, the main riser on a down-feed system from Column *B*, the dry return from Column *O*, and the wet return from Column *N*.

Example 3. Size the one-pipe gravity steam system shown in Fig. 1 assuming that this is all there is to the system or that the riser and run shown involves the longest run on the system.

Solution. The total length of run actually shown is 215 ft. If the equivalent length of run is taken at double this, it will amount to 430 ft, and with a total drop of ¼ lb the drop per 100 ft will be slightly less than 1⁄16 lb. It would be well in this case to use 1⁄24 lb, and this would result in the theoretical sizes indicated in Table 9. These theo-

TABLE 9. PIPE SIZES FOR ONE-PIPE UP-FEED SYSTEM SHOWN IN FIG. 1

PART OF SYSTEM	SECTION OF PIPE	RADIATION SUPPLIED (SQ FT)	THEORETICAL PIPE SIZE (INCHES)	PRACTICAL PIPE SIZE (INCHES)
Branches to Radiators	....	100	2	2
Branches to Radiators	....	50	1¼	1¼
Riser	*a* to *b*	200	2	2
Riser	*b* to *c*	300	2½	2½
Riser	*c* to *d*	400	2½	2½
Riser	*d* to *e*	500	3	3
Riser	*e* to *f*	600	3	3
Branch to Riser	*f* to *g*	600	3½	3½
Supply Main	*g* to *h*	600	3	3
Branch to Supply Main	*h* to *j*	600	2½	3
Dry Return Main	*f* to *k*	600	1¼	2
Wet Return Main	*k* to *m*	600	1	2
Wet Return Main	*m* to *n*	600	1	2
Wet Return Main	*n* to *p*	600	1	2

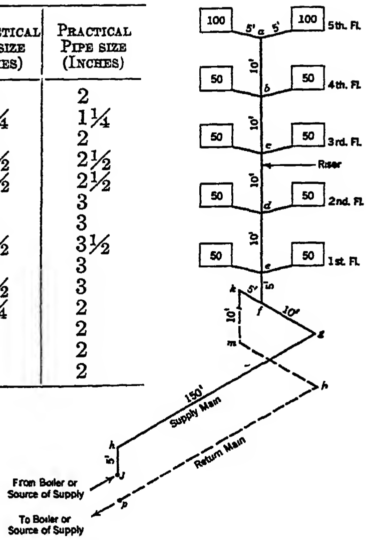


FIG. 1. RISER, SUPPLY MAIN AND RETURN MAIN OF ONE-PIPE SYSTEM

retical sizes, however, should be modified by not using a wet return less than 2 in. while the main supply, *g-h*, if from the uptake of a boiler, should be made the full size of the main, or 3 in. Also the portion of the main *k-m* should be made 2 in. if the wet return is made 2 in.

Notes on Gravity One-Pipe Air Vent Systems

1. Radiator runouts over 8 ft long should be increased one pipe size.
2. Pitch of mains should be not less than ⅛ in. in 10 ft.
3. Pitch of horizontal runouts to risers and radiators should not be less than ½ in. in 10 ft.
4. In general, it is not desirable to have a main less than 2 in. The diameter of the far end of the supply main should be not less than half its diameter at its largest part.
5. Supply mains, branches to risers, or risers, should be dripped where necessary.

SIZING TWO-PIPE GRAVITY AIR VENT SYSTEMS

The method employed in determining pipe sizes for two-pipe gravity air vent systems is similar to that described for one-pipe systems except that the steam mains never carry radiator condensation. The drop allowable per 100 ft of equivalent run is obtained by taking the equivalent length to the farthest radiator as double the actual distance, and then dividing the allowable or desired total drop by the number of hundreds of feet in the equivalent length. Thus in a system measuring 400 ft from the boiler to the farthest radiator, the approximate equivalent length of run would be 800 ft. With a total drop of ½ lb the drop per 100 ft would be $\frac{1/2}{8}$ or 1/16 lb; therefore, Column *D* would be used for all steam mains where the condensation and steam flow in the same direction. If a total drop of ¼ lb is desired, the drop per 100 ft would be 1/32 lb

and Column *B* would be used. If the total drop were to be 1 lb, the drop per 100 ft would be ⅛ lb and Column *E* would be used.

For mains and riser runouts that are not dripped, and for radiator runouts where in all three cases the condensation and steam flow in opposite directions, Column *I* should be used, while for the steam risers Column *H* should be used unless the drop per 100 ft is 1/64 lb or 1/32 lb, when Columns *B* or *C* should be substituted so as not to exceed the drop permitted.

On an overhead down-feed system the main steam riser should be sized by reference to Column *H*, but the down-feed steam risers supplying the radiators should be sized by the appropriate Columns *B* through *G*, since the condensation flows downward with the steam through them. The riser runouts, if pitched down toward the riser as they should be, are sized the same as the steam mains, and the radiator runouts are made the same as in an up-feed system.

In either up-feed or down-feed systems the returns are sized in the same manner and on the same pressure drop basis as the steam main; the return mains are taken from Columns *O*, *R*, *U*, *X*, or *AA* according to the drop used for the steam main; and the risers are sized by reading the lower part of Table 8 under the column used for the mains. The horizontal runouts from the riser to the radiator are not usually increased on the return lines although there is nothing incorrect in this practice. The same notes apply that are given for one-pipe gravity systems.

SIZING TWO-PIPE VAPOR SYSTEMS

While many manufacturers of patented vapor heating accessories have their own schedules for pipe sizing, an inspection of these sizing tables indicates that in general as small a drop as possible is recommended. The reasons for this are: (1) to have the condensation return to the boiler by gravity, (2) to obtain a more uniform distribution of steam throughout the system, (3) because with large variation in pressure the value of graduated valves on radiators is destroyed.

For small vapor systems where the equivalent length of run does not exceed 200 ft, it is recommended that the main and any runouts to risers that may be dripped should be sized from Column *D*, while riser runouts not dripped and radiator runouts should employ Column *I*. The up-feed steam risers should be taken from Column *H*. On the returns, the risers should be sized from Column *U* (lower portion) and the mains from Column *U* (upper portion). It should again be noted that the pressure drop in the steam side of the system is kept the same as on the return side except where the flow in the riser is concerned.

On a down-feed system the main vertical riser should be sized from Column *H*, but the down-feed risers can be taken from Column *D* although it so happens that the values in Columns *D* and *H* correspond. This will not hold true in larger systems.

For vapor systems over 200 ft of equivalent length, the drop should not exceed ⅛ lb to ¼ lb, if possible. Thus, for a 400 ft equivalent run the drop per 100 ft should be not over ⅛ lb divided by 4, or 1/32 lb. In this case the steam mains would be sized from Column *B*; the radiator and undripped riser runouts from Column *I*; the risers from Column *B*,

because Column *H* gives a drop in excess of $\frac{1}{32}$ lb. On a down-feed system, Column *B* would have to be used for both the main riser and the smaller risers feeding the radiators in order not to increase the drop over $\frac{1}{32}$ lb. The return risers would be sized from the lower portion of Column *O* and the dry return main from the upper portion of the same column, while any wet returns would be sized from Column *N*. The same pressure drop is applied on both the steam and the return sides of the system.

Notes on Vapor Systems

1. Radiator runouts over 8 ft long should be increased one pipe size.

2. Pitch of mains should be not less than $\frac{1}{8}$ in. in 10 ft.

3. Pitch of horizontal runouts to risers and radiators should be not less than in. $\frac{1}{2}$ in 10 ft.

4. In general it is not desirable to have a supply main smaller than 2 in., and when the supply main is 3 in. or over at the boiler or pressure reducing valve it should be not less than $2\frac{1}{2}$ in. at the far end.

5. When necessary, supply main, supply risers, or branches to supply risers should be dripped separately into a wet return. The drip for a vapor system may be connected into the dry return through a thermostatic drip trap.

SIZING VACUUM SYSTEMS

Vacuum systems are usually employed in large installations and have total drops varying from $\frac{1}{4}$ to $\frac{1}{2}$ lb. Systems where the maximum equivalent length does not exceed 200 ft preferably employ the smaller pressure drop while systems over 200 ft equivalent length of run more frequently go to the higher drop, owing to the relatively greater saving in pipe sizes. For example, a system with 1200 ft longest equivalent length of run would employ a drop per 100 ft of $\frac{1}{2}$ lb divided by 12, or $\frac{1}{24}$ lb. In this case the steam main would be sized from Column *C*, and the risers also from Column *C* (Column *H* could be used as far as critical velocity is concerned but the drop would exceed the limit of $\frac{1}{24}$ lb). Riser runouts, if dripped, would use Column *C* but if undripped would use Column *I*; radiator runouts, Column *I*; return risers, lower part of Column *S*; return runouts to radiators, one pipe size larger than the radiator trap connections.

Notes on Vacuum Systems

1. It is not generally considered good practice to exceed $\frac{1}{8}$ lb drop per 100 ft of equivalent run nor to exceed 1 lb total pressure drop in any system.

2. Radiator runouts over 8 ft long should be increased one pipe size.

3. Pitch of mains should be not less than $\frac{1}{8}$ in. in 10 ft.

4. Pitch of horizontal runouts to risers and radiators should be not less than $\frac{1}{2}$ in. in 10 ft.

5. In general it is not considered desirable to have a supply main smaller than 2 in. When the supply main is 3 in. or over, at the boiler or pressure reducing valve, it should be not less than $2\frac{1}{2}$ in. at the far end.

6. When necessary, the supply main, supply riser, or branch to a supply riser should be dripped separately through a thermostatic trap into the vacuum return. A connection should not be made between the steam and return sides of a vacuum system without interposing a thermostatic trap to prevent the steam from entering the return line.

7. Lifts should be avoided if possible, but when they cannot be eliminated they should be made in the manner described in Chapter 31 under *Up-Feed Vacuum Systems*.

SIZING ATMOSPHERIC SYSTEMS

The sizing of the supply and return piping on atmospheric systems is practically identical with the sizing used for vacuum systems and the same notes apply, except that no lift can be made in the return line.

SUB-ATMOSPHERIC SYSTEM SIZING

Any properly pitched, correctly sized vacuum system without a lift may be used as a sub-atmospheric system when the proper equipment is substituted for the ordinary vacuum pump, traps, and controls. On new systems manufacturers usually recommend a drop on the steam line of between ¼ and ⅜ lb for the total run, and suggest adding 25 ft to the total equivalent length of run to insure that the steam gets through to the last radiator.

The same notes apply to these systems as for vacuum systems, except that no lifts can be made in the returns.

SIZING ORIFICE SYSTEMS

The orifice systems can be operated with any piping system suitable for vacuum operation according to experienced designers. Because these systems vary considerably in detail, it is advisable to consult the manufacturer of the particular system contemplated for recommendations.

The same notes apply to these systems as to vacuum systems, except that lifts cannot be made in the returns of orifice systems if a vacuum pump is used.

HIGH PRESSURE STEAM

When steam heating systems are supplied with steam from a high pressure plant, one or more pressure-reducing valves are used to bring the pressure down to that required by the heating system. It has been considered good practice to make the pressure reductions in steps not to exceed 50 lb in each case. For example, in reducing from 100 lb gage to 2 lb gage, two pressure reducing valves would be used, the first reducing the pressure from 100 lb gage to 50 lb and the second reducing the pressure from 50 lb gage to 2 lb gage. Valves are available that will reduce 100 lb in one step, and it is questionable whether two valves are now required for initial pressures of 150 lb or less.

The pressure-reducing valve, or pressure-regulator as it is sometimes termed, has ratings which vary 200 to 400 per cent. Some of these ratings are based on arbitrary steam velocities through the valve of 5,000 to 10,000 fpm and it is assumed that the valve when wide open has the same area as the pipe on the inlet opening of the valve. It is well known that steam flowing through an orifice increases its velocity until the pressure on the outlet side is reduced to 58 per cent of the absolute pressure on the inlet side and that with further reduction of pressure on the outlet side little change in velocity will be obtained. As practically all pressure-reducing valves used for steam heating work lower the steam pressure to less than 58 per cent of the inlet pressures, only the maximum velocity through such valves need be considered. If it is assumed that the valve, when fully open, has an area equal to that of the inlet pipe size,

TABLE 10. CAPACITIES OF PRESSURE REDUCING VALVES
(100 LB GAGE DOWN TO ANY PRESSURE—52 LB OR LESS)

INLET NOMINAL PIPE DIAMETER (INCHES)	POUNDS STEAM PER HOUR AT 100 LB GAGE	EQUIVALENT DIRECT RADIATION SQ FT AT ¼ LB	EQUIVALENT DIRECT RADIATION SQ FT AT 1/3 LB
½	866	3,464	2,598
¾	1,576	6,304	4,728
1	2,459	9,836	7,377
1¼	4,263	17,052	12,689
1½	5,808	23,232	17,424
2	9,564	38,256	28,692
2½	13,623	54,492	40,869
3	21.041	84,104	63,123
3½	28,213	112,852	84,039
4	36,285	145,140	108,855
5	56,971	227,884	170,913
6	82,336	329,344	247,008

Formula:

$$\frac{A \times V \times 3600 \times 50}{144 \times 3.84} = \text{pounds per hour passed by orifice.}$$

where

A = area of inlet pipe in square inches.
V = velocity of steam through orifice (approximately 870 fps).
50 = 70 per cent efficiency of orifice less 20 per cent for factor of safety.
144 = square inches in 1 sq ft.
3600 = seconds in one hour.
3.8 = cubic feet per pound at 100 lb gage.

that the steam is flowing into a pressure less than 58 per cent of the initial pressure, that the orifice efficiency is approximately 70 per cent, and that 20 per cent more is allowed for a factor of safety, then the pressure reducing valves will have the working capacities shown in Table 10. If the valve, when fully open, does not give an orifice area equal to that of the pipe on the inlet side, then the capacities will be proportional to the percentage of opening secured, taking the pipe area as 100 per cent.

Most exact regulation of pressure on steam heating systems is secured from diaphragm-operated valves controlled by a pilot line from the low pressure pipe, taken off the low pressure main at least 15 ft from the reducing valve. The reducing valves operating on the proportional-reduction principle will give a variation of steam pressure on the low pressure side if the initial pressure varies between considerable limits. The so-called dead-end valve is used for reduced pressures where the line has not sufficient condensing capacity at all times to condense the leakage that might occur with the ordinary valve. Single-disc valves do not give as close regulation as double-disc valves, but the single disc is preferable where dead-end valves are necessary, such as on short runs to thermostatically controlled hot water heaters, central fan heating units and unit heaters.

The correct installation (Fig. 2) of a pressure-reducing valve includes a pressure-reducing valve with a gate valve on each side, a by-pass controlled by a globe valve, a pressure gage on the low pressure side, and a safety valve on the low pressure main at some point, usually within a reasonable distance of the pressure-reducing valve. Pressure-reducing valves should have expanded outlets for sizes greater than 2 in. Where the steam main is of still larger diameter than the expanded outlet, and in

cases where straight valves are used, an increaser is placed close against the outlet of the valve to reduce the velocity immediately after passing through the valve. Strainers are recommended on the inlets of all pressure-reducing valves. A pressure gage may be located on the high-pressure line near the valve if desired.

Owing to the large variation in steam demand on the average heating system, it is generally advisable to use two pressure-reducing valves connected in parallel. One valve should be large enough for the maximum load and the other should have a diameter approximately half that of the first. The smaller valve can be used most of the time, for it will give much better regulation than the larger one on light or normal loads.

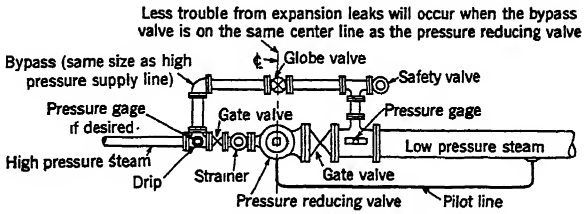


FIG. 2. TYPICAL PRESSURE REDUCING VALVE INSTALLATION

Control Valves

Gate valves are recommended in all cases where service demands that the valve be either entirely open or entirely closed, but they should never be used for throttling. Angle globe valves and straight globe valves should be used for throttling, as done on by-passes around pressure reducing valves or on by-passes around traps.

EXPANSION IN STEAM AND RETURN LINES

Because all steam and return lines expand and contract with changes in temperature, provision should be made for such movement. The expansion in steam supply pipes is normally taken at $1\frac{1}{4}$ to $1\frac{1}{2}$ in. per 100 ft and in return lines at one-half or two-thirds of this amount. It may be calculated accurately if the temperature rise and fall can be determined with reasonable certainty (Table 3, Chapter 34). The temperature at the time of erection often has a greater expansion effect on piping than the temperature in the building after it has been put into service.

Expansion may be taken care of by any, or all, of three different methods, namely, (1) the spring in the pipe including offsets and expansion bends, (2) the turning of the pipe on its threads and swing joints, and (3) the use of expansion joints.

By the first scheme, which is the most popular method where space permits, the pipe is offset, or *broken*, around rooms or corners, and is hung so that the spring in the pipe at right angles to the expansion movement is sufficient to absorb the expansion. If conditions do not lend themselves to this treatment, regular expansion bends of the *U* or offset type may be used. In tight places such as pipe tunnels the expansion joint is preferable.

On riser runouts and radiator runouts the swing joint is used almost without exception. On high vertical risers the pipes may be reversed every five to ten stories; that is, the supply is carried over to the adjacent return riser location and the return riser is run over to the former supply riser location, thus making horizontal offsets in each line. Corrugated copper expansion joints also are used on risers but must be made accessible in case future replacement becomes necessary.

EXPANSION BENDS

The calculation of the distance required for offsets and the size of expansion bends necessary to absorb a given amount of expansion leads into complicated formulas and is a subject of controversy. It seems to have been demonstrated, however, that the shape of the bend, the radius used, the relative amounts of straight and curved pipe in a bend, and the

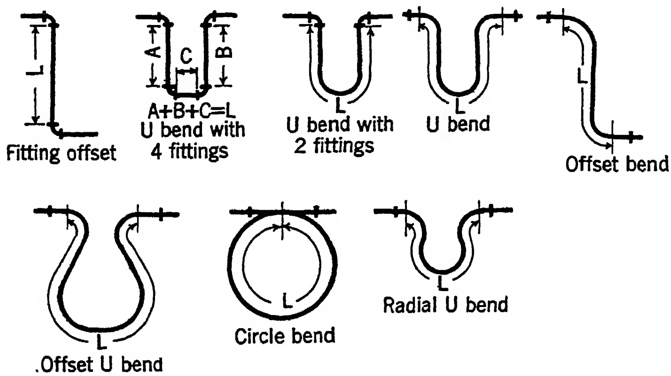


FIG. 3. MEASUREMENT OF L ON VARIOUS PIPE BENDS AND OFFSETS FOR ABSORBING EXPANSION

type of bend have little bearing on the amount of expansion for which they will safely provide. The size, weight and material of the pipe and the length of all of the pipe in the bend, or even in the offset, have a bearing on its capacity to absorb expansion without straining the pipe material beyond the safe working stress. In Fig. 3 typical pipe bends and offsets for absorbing expansion are shown. The lengths L are those which are used in determining the stress in the pipe.

Fig. 4 shows a set of curves for standard weight steel pipe bends from which the approximate amount of pipe L (Fig. 3) for each pipe size may be determined from the amount of expansion movement that must be absorbed. These curves are such that the maximum fiber stress in any part of the bend will not be over 16,000 lb per square inch. Since 12,000 lb per square inch is considered to be a maximum working fiber stress in wrought iron pipe, an additional 33⅓ per cent must be added to the length of this type of pipe.

The amount of expansion can be doubled for a bend if the bend is cold sprung for one-half of the expansion movement. In other words, if the bend is erected with the main pipe cut short one-half of the expected expansion and the bend is then sprung open to meet the shortened pipe,

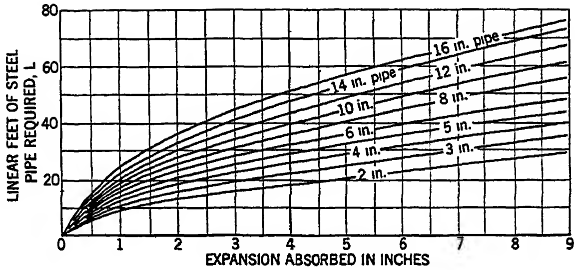


FIG. 4. CURVES GIVING LENGTH L OF BEND OF OFFSET NECESSARY TO ABSORB EXPANSION (WITHOUT COLD SPRING)

the expansion in the main will first allow the bend to go back to its neutral point and then will compress the bend an equal distance beyond the neutral point, thus securing a doubled capacity. Generally only a portion of the cold spring is considered as being effective owing to the difficulties of erecting the bends with sufficient exactitude in the length of the main line and the difficulty of cold springing.

PIPING CONNECTIONS AND DETAILS

Piping connections may be classified into two groups: *first,* those suitable for any system of steam heating; *second,* those devised for certain systems which cannot be satisfactorily applied to any other type. There are also various details that apply to piping on the steam side which cannot be used on the returns. An installation that is designed and sized correctly and installed with care may be rendered defective by the use of improper connections, such as runouts that do not allow for expansion, thermostatic traps unprotected from scale, pressure-reducing valves without strainers, and lack of drips at required points.

BOILER CONNECTIONS

Supply

Boiler headers and connections have the largest sizes of pipe used in a system. Cast-iron, horizontal-type, low pressure heating boilers usually have several tapped outlets in the top, the manufacturers recommending their use in order to reduce the velocity of the steam in the vertical uptakes from the boiler and to permit entrained water to return to the boiler instead of being carried over into the steam main where it must be cared for by dripping. Steel heating boilers usually are equipped with only one steam outlet but many engineers believe that better results are obtained by specifying that such boilers have two. The second outlet, usually located 3 or 4 ft back of the regular one, reduces the velocity 50 per cent in the steam uptake.

Fig. 5 shows a type of boiler connection that was used for many years and one with which some boilers are now piped. The uptakes are carried as high as possible, turned horizontally and run out to the side of the boiler and then are connected together into the main boiler runout which drops into the top of the boiler header through a boiler stop valve. No

drips are provided on this type of runout except a very small one which is sometimes installed on the boiler side of the stop valve. Fig. 6 shows a type of boiler connection which is regarded as superior to that shown in Fig. 5 and which is the type illustrated in the system diagrams in Chapter 31. This type is similar to that shown in Fig. 5 except that the horizontal branches from the uptakes are connected into the main boiler runout, and the steam is carried toward the rear of the boiler. The branch to the building or boiler header is taken off *behind* the last horizontal boiler connection. At the rear end of this main runout, a large size drip, or balance pipe, is dropped down into the boiler return, or into the top of the Hartford Loop, which is described in a following paragraph. As a result, any water carried over from the boiler follows the direction of steam flow

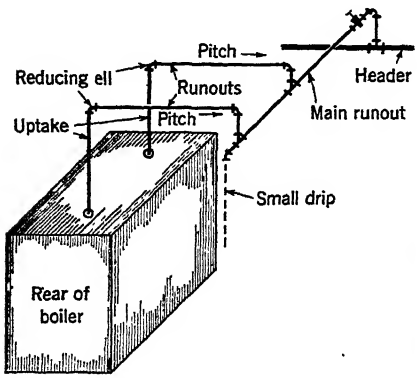


Fig. 5. Old Style Standard Boiler Connections

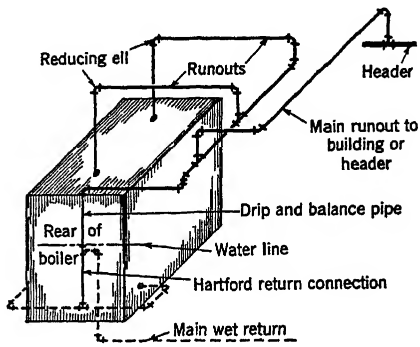


Fig. 6. Approved Method of Boiler Connections

toward the rear and is discharged into the rear drip, or balance pipe, without being carried over into the system.

Return

Cast-iron boilers are generally provided with return tappings on both sides, but steel boilers often are equipped with only one return tapping. A boiler with side return tappings will usually have a more effective circulation if both tappings are used. Check valves generally should not be used on the return connection to steam heating boilers because they are not always dependable inasmuch as a small piece of scale or dirt lodged on the seat will hold the tongue open and make the check useless. These valves also offer a certain amount of resistance to the returns coming back to the boiler, and in gravity systems will raise the water line in the far end of the wet return several inches[3]. However, if check valves are omitted and the steam pressure is raised with the boiler steam valve closed, the water in the boiler will be blown out into the return system with the accompanying danger of boiler damage. These objections are largely overcome with the Hartford return connection.

[3]See method of calculating height above water line for gravity one-pipe systems in Chapter 31.

Hartford Return Connection

In order to prevent the boiler from losing its water under any circumstances, the use of the Hartford Connection, or the Underwriters Loop, is recommended. Fig. 7 shows this connection for both single boiler and two-boiler installations. By balancing the column of water in the loop against the steam pressure, the water cannot be blown out of the loop whatever the relative pressure conditions in the boiler, steam lines, or return lines. This balancing is done by raising the return to approximately the normal water line of the boiler, looping it back to the boiler inlet and connecting the top of this loop by means of a balance pipe with the steam runout from the boiler. It is important that this balance pipe be connected into the boiler steam line on the boiler side of all valves.

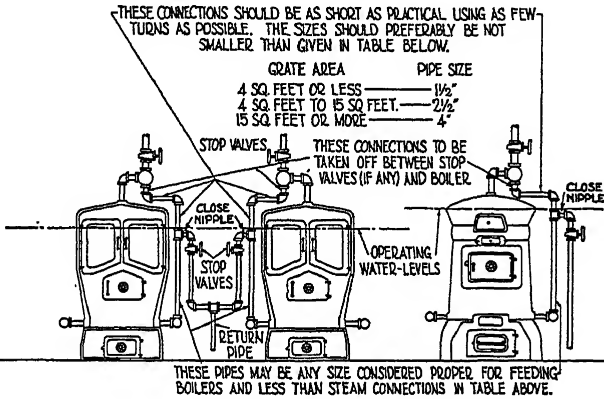


Fig. 7. The Hartford Return Connection

Theoretically, the top of the loop should be at the normal boiler water line but since this installation often causes trouble from water hammer in the top of the loop, this top is usually made 2 in. below the normal boiler water line to keep the horizontal pipe at the top submerged under all normal conditions. It is important that this top of the loop be made with the shortest possible horizontal pipe, a close nipple being employed.

Sizing Boiler Connections

Little authentic information is available on the sizing of boiler runouts and steam headers. Although many engineers prefer an enlarged steam header to serve as additional steam storage space, there ordinarily is no sudden demand for steam in a steam heating system except during the heating-up period, at which time a large steam header is a disadvantage rather than an advantage. The boiler header may be sized by first computing the maximum load that must be carried by any portion of the header under any conceivable method of operation and then applying the same schedule of pipe sizing to the header as is used on the steam mains for the building. The horizontal runouts from the boiler, or boilers, may be sized by calculating the heaviest load that will be placed on the boiler at any time, and sizing the runout on the same basis as the building mains. The difference in size between the vertical uptakes from the

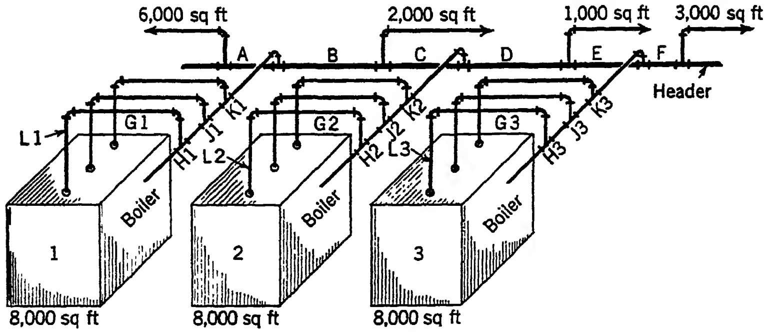


FIG. 8. BOILER STEAM HEADER AND CONNECTIONS

boiler and the horizontal main or runout is compensated for by the use of reducing ells (Figs. 5 and 6).

The following example illustrates the sizing of the boiler connections shown in Fig. 8.

Example 4. Determine the size of boiler steam header and connections (Fig. 8) if there are three boilers, two to carry 50 per cent of the load each, and the third to be used as a spare. The steam mains are based on ⅛ lb drop per 100 sq ft of equivalent direct radiation (EDR).

Solution:

Size of Boiler Header

When Operating on Boilers	Load on Various Portions of Header						Maximum Load
	A	*B*	*C*	*D*	*E*	*F*	
Nos. 1 and 2	6000	0	2000	4000	3000	3000	6000
Nos. 2 and 3	6000	6000	8000	2000	3000	3000	8000
Nos. 3 and 1	6000	0	2000	2000	3000	3000	6000
Max. Load	6000	6000	8000	4000	3000	3000	8000

8000 sq ft @ ⅛ lb per 100 ft = 6 in. main. (See Table 7).

Size of Boiler Runouts

The three runouts

$G_1, G_2, G_3 = \frac{8000}{3} = 2667$ sq ft each @ ⅛ lb per 100 ft = 4 in. pipe.

$H_1, H_2, H_3 = 2667$ sq ft each @ ⅛ lb per 100 ft = 4 in. pipe[4] (See Table 7).

$J_1, J_2, J_3 = 5333$ sq ft each @ ⅛ lb per 100 ft = 5 in. pipe[4] (See Table 7).

$K_1, K_2, K_3 = 8000$ sq ft each @ ⅛ lb per 100 ft = 6 in. pipe[4] (See Table 7).

The uptakes from the boiler probably would be 6 in. pipe with a 6 in. × 4 in. reducing ell at top.

Return connections to boilers in gravity systems are made the same size as the return main itself. Where the return is split and connected to

[4]*Note.*—As K_1, K_2, K_3 all carry 8000 sq ft and are 6 in. pipe, the whole runout including J_1, J_2 and J_3 would be made 6 in. pipe, also.

two tappings on the same boiler, both connections are made the full size of the return line. Where two or more boilers are in use, the return to each may be sized to carry the full amount of return for the maximum load which that boiler will be required to carry. Where two boilers are used, one of them being a spare, the full size of the return main would be carried to each boiler, but if three boilers are installed, with one spare, the return line to each boiler would require only half of the capacity of the entire system, or, if the boiler capacity were more than one-half the entire system load, the return would be sized on the basis of the maximum boiler capacity. As the return piping around the boiler is usually small and short, it should not be sized to the minimum.

With returns pumped from a vacuum or receiver return pump, the size of the line may be calculated from the water rate on the pump discharge when it is operating, and the line sized for a very small pressure drop, the size being obtained from the Chart for Friction Losses for Various Rates of Flow of Water, Fig. 3, Chapter 39. The relative boiler loads should be considered, as in the case of gravity return connections.

Radiator Connections

Radiator connections are important on account of the number of repetitions which occur in every heating installation. They must be properly pitched and they must be arranged to allow not only for movement in the riser but, in frame buildings, for the shrinkage of the building. In a three story building this sometimes amounts to 1 in. or more. The simplest connection is that for the one-pipe system where only one radiator connection is necessary. Where the radiator runouts are located on the ceiling or under the floor, sufficient space usually is available to make a good swing joint with plenty of pitch, but where the runouts must come above the floor the vertical space is small and the runouts can project out into the room only a short distance. Fig. 9 illustrates two satisfactory methods of making runouts on a one-pipe gravity air vent system of either the up-feed or down-feed type, the runout below the floor being indicated in full lines and the runout above the floor in dotted lines. Sometimes it is necessary to set a radiator on pedestals, or to use high legs, in order to obtain sufficient vertical distance to accommodate above-the-floor runouts. Particular attention must be given to the riser expansion as it will raise the runout and thereby reduce the pitch.

Similar connections for a two-pipe system of the gravity air vent type are illustrated in Fig. 10 for the old steam type radiator. If the water type is used, the supply tapping is at the top instead of at the bottom, the runouts otherwise remaining as shown in Fig. 10. A satisfactory type of radiator connection for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of both the up-feed and down-feed types is shown in Fig. 11.

While short radiators, not exceeding 8 to 10 sections, may be supplied and returned from the same end as indicated in Fig. 12, the top-an-bottom-opposite-end method is to be preferred in all cases where it can be used. On down-feed systems of the atmospheric, vapor, vacuum, sub-atmospheric, and orifice types, the bottom of the supply riser must be dripped into the return somewhat as illustrated in Fig. 13. On up-feed systems of the vapor and atmospheric types, where radiators in the

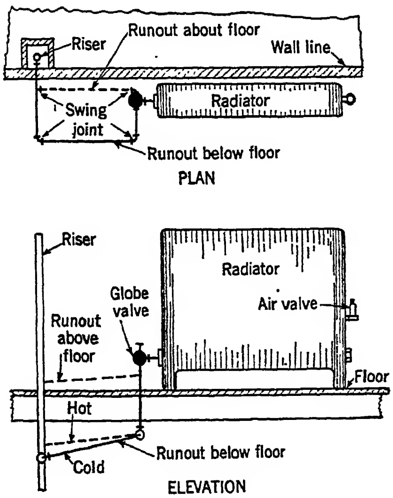


FIG. 9. TYPICAL ONE-PIPE RADIATOR CONNECTIONS (UP-FEED OR DOWN-FEED)

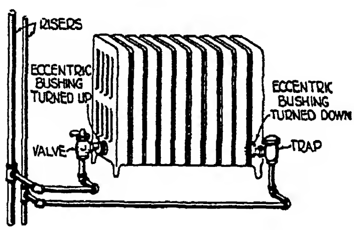


FIG. 10. CONNECTIONS TO STEAM-TYPE RADIATOR FOR TWO-PIPE GRAVITY SYSTEM, UP-FEED OR DOWN-FEED

Note.—Steam-type radiators should not be used on any except gravity one-pipe and gravity two-pipe systems.

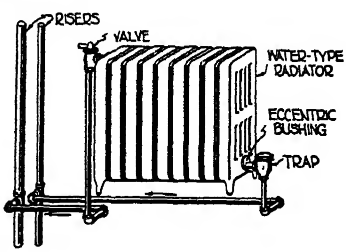


FIG. 11. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS FROM UP OR DOWN-FEED RISERS

Note.—Suitable for up-feed or down-feed atnospheric, vapor, vacuum, sub-atmospheric and orifice systems.

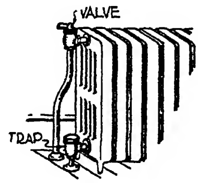


FIG. 12. TOP AND BOTTOM RADIATOR CONNECTIONS FROM UP- OR DOWN-FEED RISERS. (NOT TO EXCEED 8 TO 10 SECTIONS).

Note.—Suitable for up-feed or down-feed atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems. Opposite end connections always preferable.

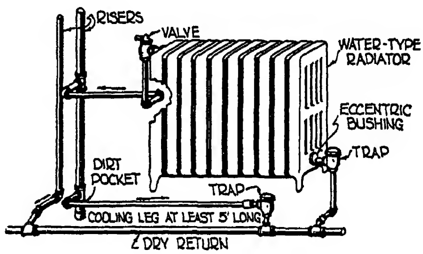


FIG. 13. TOP AND BOTTOM OPPOSITE END RADIATOR CONNECTIONS WITH HEEL OF DOWN-FEED RISER DRIPPED INTO DRY RETURN

Note.—Suitable for down-feed only. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

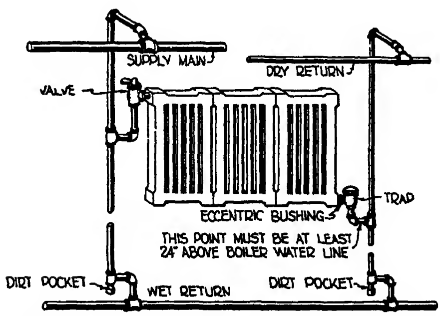


FIG. 14. CONNECTIONS TO RADIATOR HUNG ON WALL

Note.—For up-feed with radiators below level of steam main. For atmospheric and vapor systems. Not suitable for vacuum, sub-atmospheric, or orifice systems.

basement are located below the level of the steam main, the drop to the radiator is dripped into the wet return and an air line is used to vent the return radiator connection into an overhead return line, as illustrated in Fig. 14. When the radiator stands on the floor below the main, the drip on the steam branch down to the radiator may be omitted if an overhead valve, as shown in Fig. 15, is used. This method is also suitable for vacuum, sub-atmospheric, and orifice systems.

Convector Connections

Convectors often are installed without control valves, a damper being used to shut off the flow of air to retard the heat transfer from the convector even though it is still supplied with steam. The piping connections for a convector with the inlet and outlet at the same end are shown in Fig. 16. There is no valve on the steam side but there is a thermostatic trap on the return. The damper for control is shown immediately above the convector. This piping is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems of the up-feed type. A similar unit with connections on opposite ends and suitable for the same systems is shown in Fig. 17. This unit has no damper but requires a valve on the steam connection for control. When valves must be located so as to be accessible from the supply air grille, the arrangement usually takes the form indicated in Fig. 18. Convectors with damper control, installed in cabinets or under window sills, usually are connected as shown in Fig. 19. A convector located in the basement and supplying air to a room on the floor above may be piped as pictured in Fig. 20 for all systems except gravity one-pipe or two-pipe systems.

Vapor systems with heating units in the basement where the returns are wet would be treated as in Fig. 21. Similar heating units where a dry return is available would be connected as shown in Fig. 22. If the dry return were on a vacuum, atmospheric, sub-atmospheric or orifice system, the treatment would be identical.

Pipe Coil Connections

Pipe coils, unless coupled in a correct manner, often give trouble from short circuiting and poor circulation. The method of connecting shown in Fig. 23 is suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

Indirect Air Heater Connections

Heating units for central fan systems have simple connections on the steam side. The steam main is carried into the fan room and has a single branch tapped off for each row of heating units. Each of these main branches is split into as many connections as need be made to each row, governed by the number of stacks and the width of the stacks. Each stack must have at least one steam connection, and wide stacks are more evenly heated with two steam connections, one at each end.

The piping shown in Fig. 24 is for small stacks and has the steam connected at only one end. On the return side all of the returns are collected together through check valves and are passed through blast traps which are connected to the vacuum return or to an atmospheric return. The air

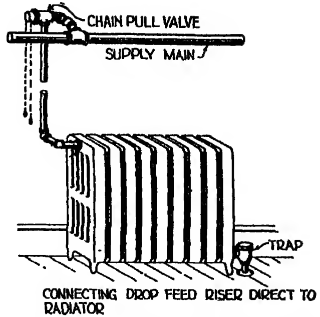


Fig. 15. Connecting Drop Feed Riser Direct to Radiator by Turning Valve on its Side

Note.—Suitable for up-feed with radiators below level of steam main. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

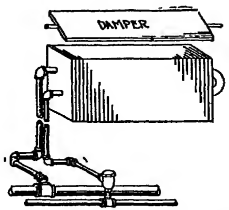


Fig. 16. Convector Connections Same End

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

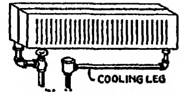


Fig. 17. Horizontal Sectional Fin-Type Heating Unit

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum sub-atmospheric, and orifice systems.

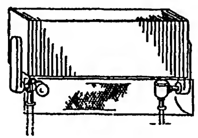


Fig. 18. Fin-Type Heating Unit Valves Behind Grille

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

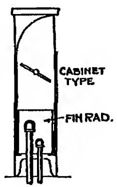


Fig. 19. Fin-Type Heating Unit Concealed in Cabinet

Note.—Suitable for any two-pipe system.

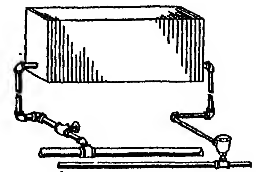


Fig. 20. Fin-Type Heating Unit with Valves in Basement

Note.—Suitable for up-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

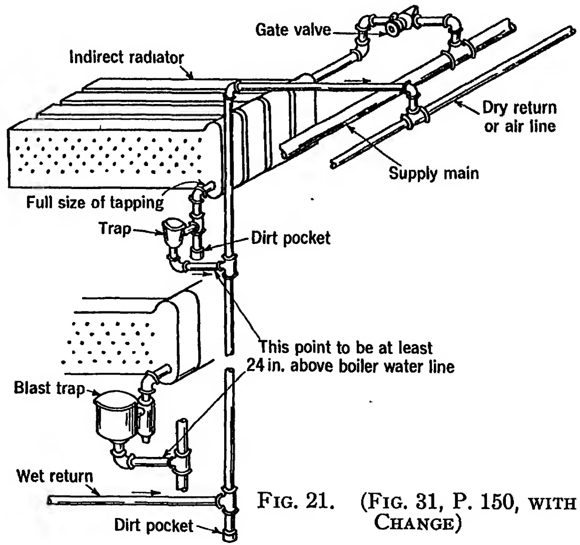


Fig. 21. (Fig. 31, P. 150, with Change)

from the stacks, in the case illustrated, passes up into a small air line and through a thermostatic trap into a line connecting into the return beyond the blast trap. It is important to use a nipple the full size of the outlet tapping on the stack and to reduce the pipe size to the normal return size required, by the use of a reducing ell, as indicated in Fig. 25.

Where the stacks contain some thirteen or more sections, an auxiliary air tapping is made to the lower portion of one of the middle sections, in the manner illustrated in Fig. 26, to prevent air collecting at this point. Thermostatic control as applied to such heating units in modern practice consists of a thermostatic valve located in each main branch from the steam line so that each valve will open or close a complete row of stacks across the entire face of the heating unit. In this case no particular attention need be paid to the method of connecting the returns, that is, they do not need to be connected in parallel with the steam connections but may be hooked together in any convenient manner. The arrangement shown in Fig. 27 is satisfactory. A detail of the arrangement where a connection is made with a stack is shown in Fig. 28. It is essential to have a check valve on each individual stack to prevent reverse flow when the thermostatic valve in the steam line closes off and a partial vacuum is produced in the stack. The end of the steam main also should be dripped as indicated in Fig. 27.

If the separate air line is used as shown in Fig. 24, the blast traps may be supplied without thermostatic by-passes but if the piping is arranged as shown in Figs. 26 or 27, the blast traps must be supplied with the thermostatic by-passes to permit the passage of the air.

PIPE SIZING FOR INDIRECT HEATING UNITS

Pipe connections and mains for indirect heating units are sized in a manner similar to radiators, but the equivalent direct radiation must be ascertained for each row of heating unit stacks and then must be divided into the number of stacks constituting that row and into the number of connections to each stack.

$$EDR = \frac{Q \times 60 \times (t_L - t_E)}{55.2 \times 240} = \frac{Q \times (t_l - t_e)}{220.8} \tag{3}$$

where

EDR = equivalent direct radiation, square feet.

Q = volume of air, cubic feet per minute.

t_e = the temperature of the air entering the row of heating units under consideration, degrees Fahrenheit.

t_l = the temperature of the air leaving the row of heating units under consideration, degrees Fahrenheit.

60 = the number of minutes in one hour.

55.2 = the number of cubic feet of air heated 1 F by 1 Btu.

240 = the number of Btu in 1 sq ft of EDR.

Example 5. Assume that the heating units shown in Fig. 27 are handling 50,000 cfm of air and that the rise in the first row is from 0 to 40 F, in the second row from 40 to 65 F, and in the third row from 65 to 80 F. What is the load in EDR on each supply and return connection?

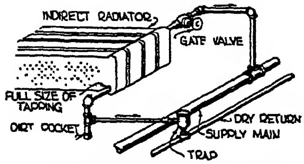


FIG. 22. TYPICAL PIPING CONNECTIONS TO INDIRECT RADIATORS WITH DRY RETURN

Note.—Suitable for atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

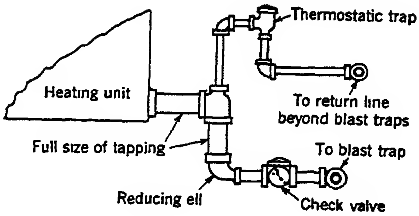


FIG. 25. HEATING UNIT RETURN CONNECTION WITH SEPARATE AIR LINE

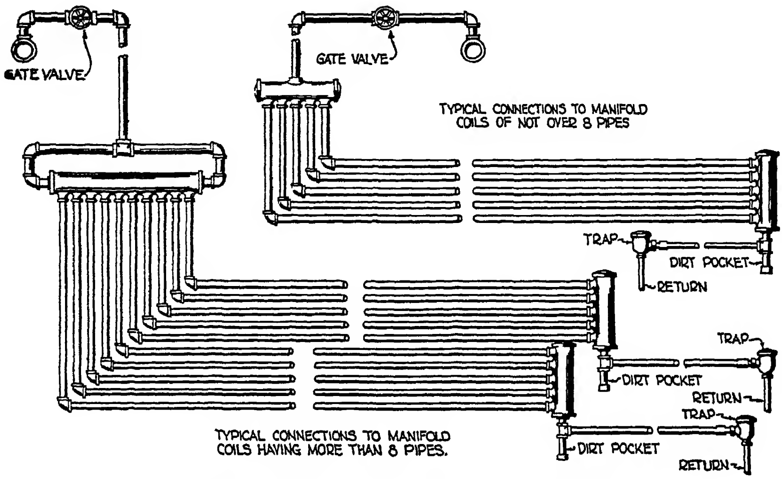


FIG. 23. TYPICAL PIPE COIL CONNECTIONS

Note.—Suitable for up-feed or down-feed. For atmospheric, vapor, vacuum, sub-atmospheric, and orifice systems.

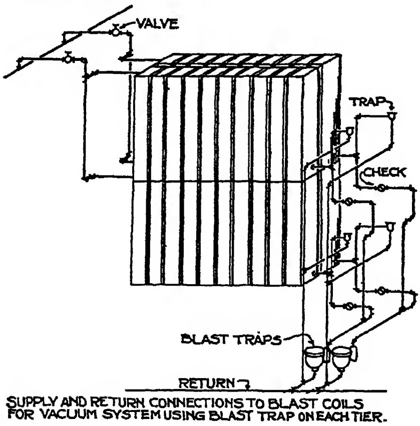


FIG. 24. CONNECTIONS FOR HEATING UNITS OF CENTRAL FAN SYSTEMS

Note.—Suitable for atmospheric and vacuum systems.

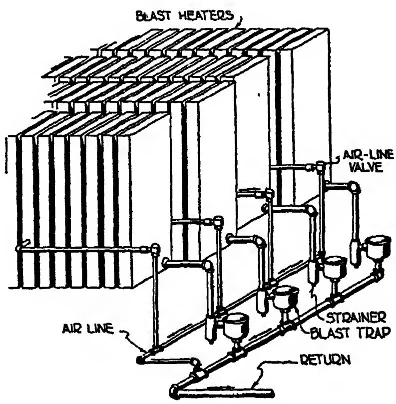


FIG. 26. TYPICAL CONNECTIONS TO CENTRAL FAN SYSTEM HEATING UNITS EXCEEDING 12 SECTIONS

Note.—Suitable for vacuum and atmospheric systems.

Solution. For the C row,

$$R = \frac{50{,}000 \times (40 - 0)}{220.8} = 9058 \text{ sq ft.}$$

For the B row,

$$R = \frac{50{,}000 \times (65 - 40)}{220.8} = 5661 \text{ sq ft.}$$

For the A row,

$$R = \frac{50{,}000 \times (80 - 65)}{220.8} = 3397 \text{ sq ft.}$$

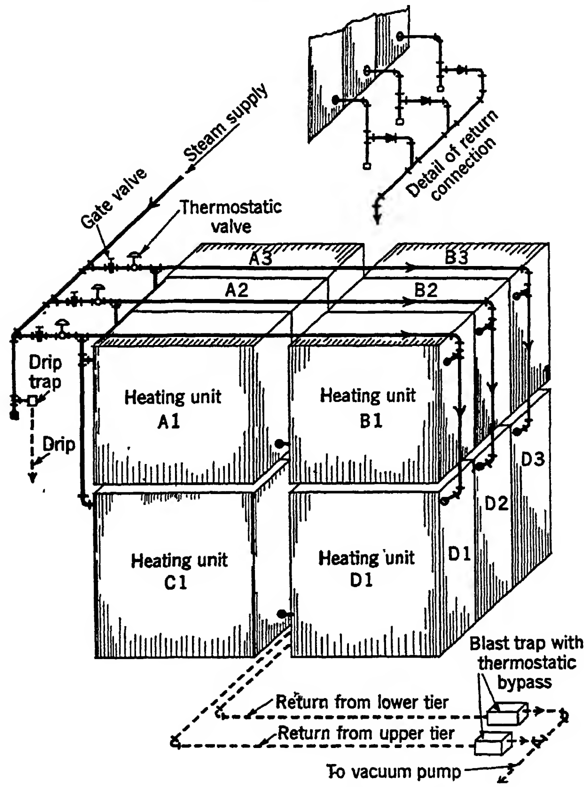


FIG. 27. TYPICAL PIPING FOR ATMOSPHERIC AND VACUUM SYSTEMS WITH THERMOSTATIC CONTROL (CENTRAL FAN SYSTEM)

Each row of heating units consists of four stacks and each stack has two connections so that the load on each stack and each connection of the stack is as follows:

Row	TOTAL LOAD (EDR)	STACK LOAD[a] (EDR)	CONNECTION LOAD[b] (EDR)
C	9058	2265	2265 or 1132
B	5661	1415	1415 or 708
A	3397	849	849 or 425

[a] ¼ of total row load.
[b] ½ of stack load if two steam connections are made; otherwise, same as stack load.

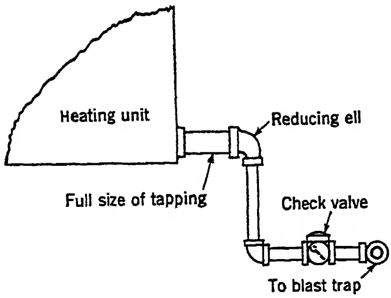


FIG. 28. HEATING UNIT RETURN CONNECTION WITHOUT SEPARATE AIR LINE (CENTRAL FAN SYSTEM)

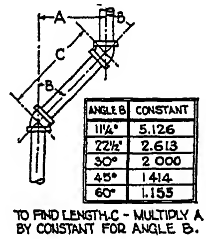


FIG. 32. CONSTANTS FOR DETERMINING PROPER LENGTH OF OFFSET PIPE

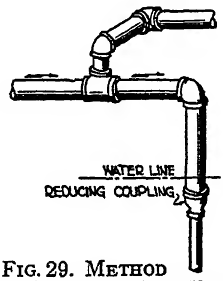


FIG. 29. METHOD OF DRIPPING MAIN WHERE IT RISES TO HIGHER LEVEL

Note.—Suitable for vapor and atmospheric systems.

FIG. 33. ACCEPTABLE AND PREFERRED METHODS OF TAKING BRANCH FROM MAIN

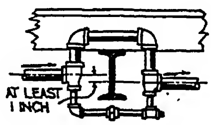


FIG. 30. LOOPING MAIN AROUND BEAM

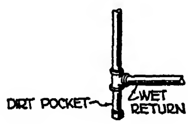


FIG. 34. DIRT POCKET CONNECTION

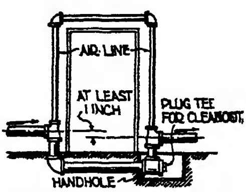


FIG. 31. LOOPING DRY RETURN MAIN AROUND OPENING

Note.—Suitable for any dry return line and any return line carrying air.

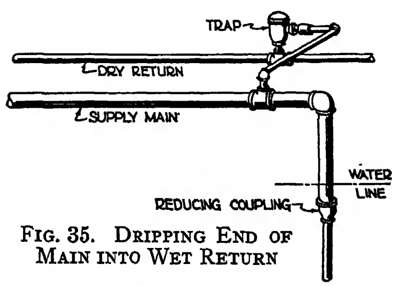


FIG. 35. DRIPPING END OF MAIN INTO WET RETURN

Note.—Suitable for vapor systems.

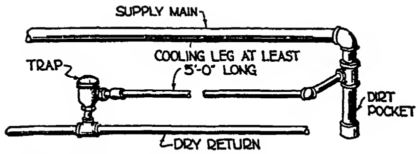


Fig. 36. Dripping End of Main into Dry Return. (A Gate Valve is Recommended at the Inlet Side of the Trap)

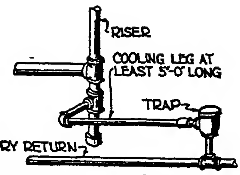


Fig. 37. Dripping Heel of Riser into Dry Return. (A Gate Valve is Recommended at the Inlet Side of the Trap)

The pipe sizes would then be based on the length of the run and the pressure drop desired, as in the case of radiators. It generally is considered desirable to place the indirect heating units on a separate system and not on supply or return lines connected to the general heating system.

DRIPPING

Any steam main in any type of steam heating system may be dropped to a lower level without dripping if the pitch is downward with the steam flow. Any steam main in any heating system can be elevated if dripped. (Fig. 29). Steam mains also may be run over obstructions without a change in level if a small pipe is carried below the obstruction to care for the condensation (Fig. 30). Return mains may be carried past doorways or other obstructions by using the scheme illustrated in Fig. 31; in vacuum systems it is well to have a gate valve in the air line.

Offsets in steam and return piping should preferably be made with 90-deg ells but occasionally fittings of other angles are used, and in such cases the length of the diagonal offset will be found as shown in Fig. 32.

Branches from steam mains in one-pipe gravity steam systems should use the *preferred connection* shown in Fig. 33, but where radiator condensation does not flow back into the main the *acceptable* method shown in the same figure may be used. This acceptable method has the advantage of giving a perfect swing joint when connected to the vertical riser or radiator connection, whereas the preferred connection does not give this swing without distorting the angle of the pipe. Runouts from the steam main are usually made about 5 ft long to provide flexibility for movement in the main.

Dirt pockets, desirable on all systems employing thermostatic traps, should be so located as to protect the traps from scale and muck which will interfere with their operation. Dirt pockets are usually made 8 in. to 12 in. deep and serve as receivers for foreign matter which otherwise would be carried into the trap. They are constructed as shown in Fig. 34.

On vapor systems where the end of the steam main is dripped down into the wet return, the air venting at the end of the main is accomplished by an air vent passing through a thermostatic trap into the dry return line as shown in Fig. 35. On vacuum systems the ends of the steam mains are dripped and vented into the return through thermostatic drip traps opening into the return line. The same method may be used in atmospheric systems. The cooling leg (Fig. 36) is for cooling the condensation

sufficiently before it reaches the trap so the trap will not be held shut by too high a temperature. On down-feed systems of atmospheric, vapor, and vacuum types, the bottom of the steam risers are dripped in the manner shown in Fig. 37.

Chapter 33

HOT WATER HEATING SYSTEMS

One- and Two-Pipe Systems, Selecting Pipe Sizes, Forced Circulation, Effect of Variations in Pipe Sizes, Gravity Circulation, Mechanical Circulation Devices, Expansion Tanks, Installation Details

A HOT water heating system is one in which water is the medium by which heat is carried through pipes from the boiler to the heating units. There are two general types, namely, *forced circulation* and *gravity circulation* systems. In the former the pressure head maintaining flow is produced mechanically, whereas in the latter the pressure head is produced by the differences in weight of the water in the flow and in the return risers.

The fundamental rule in the design of a hot water system is that the total friction and resistance head in any circuit must equal the pressure head causing the water to flow in the same circuit.

In designing a hot water heating system, it is necessary to determine:

1. The heat losses of the rooms or spaces to be heated. (See Chapter 7).
2. The size and type of boiler. (See Chapter 25).
3. The location, type, and size of heating units. (See Chapter 30).
4. The method of piping.
5. Suitable pipe sizes.
6. The type and size of circulating pump (if forced circulation).
7. The type and size of expansion tank.

The unit, a square foot of equivalent direct radiation, EDR, has been used for many years for rating purposes in both steam and hot water systems, but its use, especially in hot water systems, has always resulted in complications and confusion. It is the plan of THE GUIDE *to eventually eliminate this empirical expression and to substitute a logical unit based on the Btu. The* Mb, *the equivalent of 1000 Btu, and the* Mbh, *the equivalent of 1000 Btu per hour, which have been approved by the A.S.H.V.E., are used in this chapter on hot water systems to replace the square foot of radiation formerly used.*

ONE- AND TWO-PIPE SYSTEMS

Pipe systems may be divided into two general types, namely, *two-pipe* and *one-pipe* systems. In a two-pipe system the piping is arranged so that the water flows through only one radiator during a circuit through the system, so that all radiators are supplied with water at practically the same temperature as that in the boiler. In a one-pipe system, the water flows through more than one radiator during its circuit. In that case, the

first radiator receives the hottest water; the second radiator, somewhat cooler water; the third one, still cooler; and so on. As the temperature of the water supplied to a radiator is lowered, the size of the radiator must be increased and, consequently, the total heating surface for a one-pipe system must be greater than that for a two-pipe system for the same service.

Two-pipe systems may be divided into two classes, *direct return* systems (Fig. 1), and *reversed return* systems (Fig. 2). In a direct return system the water returns to the heater by a direct route after it has passed through its radiator and, as a result, the paths through the three radiators shown in Fig. 1 are of unequal lengths, the path through the first radiator being the shortest and that through the third radiator, the longest. In a reversed return system, the water returns to the heater by an indirect route after it has passed through the radiators, so that the paths leading through the three radiators shown in Fig. 2 are practically of equal length.

The reversed return system has an advantage over the direct return system in that it is more likely to function satisfactorily even though the

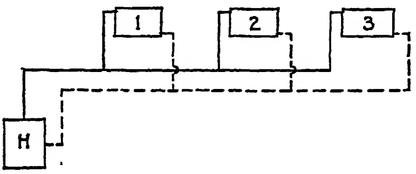


FIG. 1. A DIRECT RETURN SYSTEM

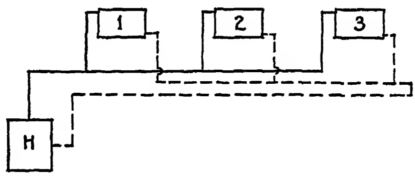


FIG. 2. A REVERSED RETURN SYSTEM

pipe system is not accurately designed. For example, if in Fig. 2 all pipes are of one size, each of the three radiators will receive approximately the same quantity of hot water because the three paths are practically of equal length, whereas in Fig. 1, if all pipes are of the same size, Radiator 1 will receive more water than the others because the path through it is shorter than those through the other radiators. As a result, Radiator 1 will be filled with water at a higher average temperature than the remaining two radiators, and will therefore dissipate more heat. To prevent this unequal distribution of heat it is necessary to throttle the paths through Radiators 1 and 2 so that the friction heads of the three paths are equal when each radiator receives its proper quantity of water.

A comparison of Fig. 1 and Fig. 2 may suggest that a reversed return system requires considerably longer mains than a direct return system. This is not always the case. For example, note the reversed return system of Fig. 3.

SELECTING PIPE SIZES

The pressure heads available in forced circulation systems are much larger than those in gravity circulation systems, consequently, higher velocities may be used in designing the system, with the result that smaller pipes may be selected and the first cost of the installation reduced. As the pipes of a heating system are reduced in size, the necessary increase in

the velocity of the water increases the cost of operating the circulating pump. There is an optimum velocity of the water in a heating system for which the sum of the cost of the system and the cost of its operation is a minimum. This velocity should be determined by calculation for the particular system under consideration.

Since the velocities in forced circulation systems are higher than those in gravity circulation systems, and since the friction heads in a heating system vary almost as the squares of the velocities, a given error in the calculation or assumption of a velocity is less important in a forced circulation system than in a gravity circulation system and, consequently, it

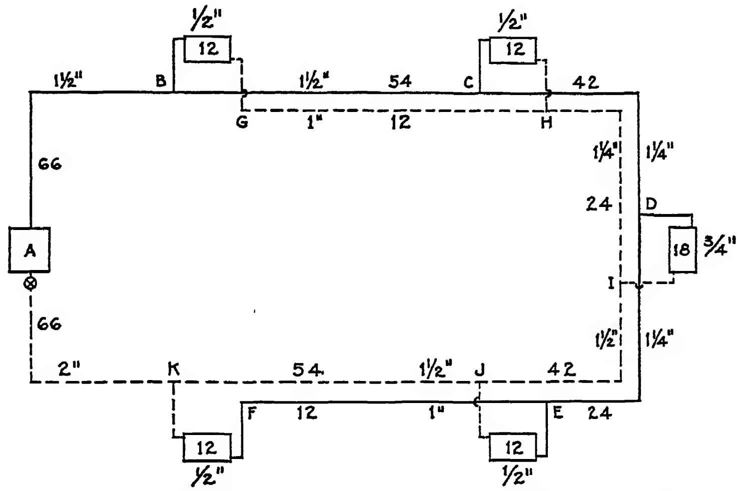


FIG. 3. A FORCED CIRCULATION REVERSED RETURN SYSTEM[a]

[a]This system could be divided into two branches. This would permit the use of smaller pipes and would produce only slight changes in the total length of the pipe. It is shown as a single system here simply to illustrate the method of determining pipe sizes by means of pipe size tables. Note that the numbers on the radiators indicate thousands of Btu per hour (Mbh) and not square feet.

is easier to design a satisfactory forced circulation system than a satisfactory gravity circulation system.

FORCED CIRCULATION

The following examples will illustrate the procedure to be followed in designing forced circulation systems:

Example 1. Assume that the paths through the five radiators shown in Fig. 3 consist each of 150 ft of mains, 5 ft of radiator connections, 1 boiler, 1 radiator, 1 radiator valve, 10 ells, and 2 tees. Design the piping for this system.

Solution. The friction heads in the boiler, radiator, valve, and tee may be expressed in terms of the friction head in one elbow according to the values given in Table 1. Having done this, each of the five circuits is taken as 155 ft of pipe and about 24 elbow equivalents. The friction head of one elbow is approximately equivalent to that in a pipe having a length equal to 25 diameters. Assuming that the average pipe size in this case will be about 1¼ in., one elbow equivalent may be placed equal to about 3 ft of pipe and the total length of the circuit equivalent to about 227 ft of pipe.

Having determined the equivalent pipe length, assume the rate at which the water is

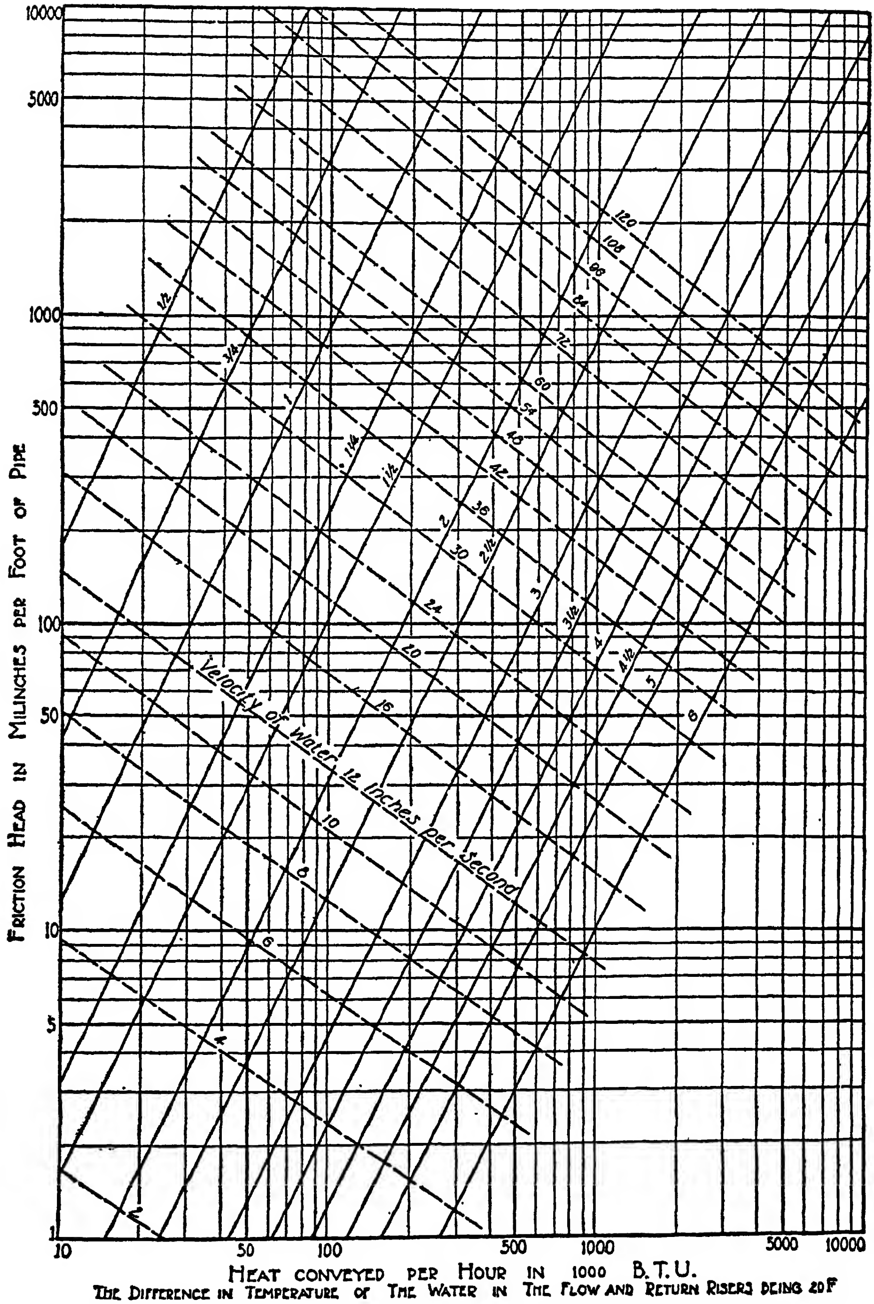


Fig. 4. Friction Heads in Pipes for a 20 deg Temperature Difference of the Water in the Flow and Return Lines

to be forced through the system. This rate may vary widely. The water may flow through the radiator so that it will cool 10 deg or 20 deg or any other reasonable number of degrees. In this case, assume a 10-deg drop. Since the system is to dissipate 66,000 Btu per hour (66 Mbh), the pump must circulate 6600 lb of water per hour or 13.8 gpm based on the actual density of water of 7.99 lb per gallon at 215 F. One gallon of water per minute at this density will deliver 9600 Btu per hour (9.6 Mbh) with a temperature drop of 20 deg.

TABLE 1. ELBOW EQUIVALENTS[a]

1 90-deg elbow	1.0
1 45-deg elbow	0.7
1 90-deg long turn elbow	0.5
1 open return bend	1.0
1 open gate valve	0.5
1 open globe valve	12.0
1 angle radiator valve	2.0
1 radiator	3.0
1 heater	3.0
1 tee	(Note[b])

[a]The loss of head in one elbow can be expressed in terms of the velocity head by the formula:

$$h = \frac{v^2}{2g} \qquad (1)$$

where

h = the loss of head in feet, v = the velocity of approach in feet per second, and $2g$ = 64.4 ft per second per second.

[b]The loss of head in tees when water is diverted at right angles through a branch of the tee varies with the per cent diverted. When the water diverted is less than 60 per cent of that approaching the tee, the loss of head, in elbow equivalents, may be expressed as follows:

$$h_e = \frac{v_1^2}{v_2^2} \qquad (2)$$

where

h_e = the loss of head in elbow equivalents, v_1 = the velocity of approach, v_2 = the velocity of water diverted at right angles.

Values in elbow equivalents for the most common percentages of water diverted in a 1x1x1-in. tee are as follows:

25%	16.0
33%	9.0
50%	4.0
100%	1.8

For other percentages the approximate values may be secured by interpolation. When the water is diverted from the tee into a smaller size branch, as in a 1x1x¾-in. tee, approximate values may be secured by means of Formula 2.

The next step in the design is to assume the velocity at which the water is to circulate through the system. This also may vary materially. As the velocity is increased, the sizes of the pipes and the cost of the system are decreased, but the cost of operating the circulating pump is increased. The designing engineer should make a careful study to determine the velocity which will produce the most economical installation for the particular case in hand. In this case, assume a velocity of about 1½ fps for a 1¼-in. pipe.

Reference to Fig. 4 shows that for a 1¼-in. pipe and a velocity of 18 in. per second, the friction head is about 100 milinches per foot, or about 2 ft for a circuit of 227 ft, if the pipe sizes for that circuit are chosen so that the average friction head is about 100 milinches per foot of pipe.

The pipe sizes may now be selected from Fig. 4 by making allowance for the fact that Fig. 4 is based on a temperature drop of 20 deg and that the system to be designed is to have a temperature drop of only 10 deg as follows: Sections *AB* and *KA* carry 66,000 Btu per hour (6.6 Mbh) with a temperature drop of 10 deg; if the temperature drop were 20 deg these sections would, with the same velocity and the same friction head, carry 132,000 Btu per hour (132 Mbh). Hence, refer to Fig. 4 for 132,000 Btu and a unit friction head of 100 milinches, and note that the correct size would be about halfway between a 1½-in. and a 2-in. pipe. Therefore, select a 1½-in. pipe for Section *AB* and

a 2-in. pipe for Section KA. The pipe sizes for the remaining eight sections and for the radiator connections can be selected in the same manner and recorded on the pipe diagram as shown.

The circulating pump for the system should be one which has its highest efficiency when it is delivering 13.8 gpm against a head of 2 ft.

If a number of heating systems are to be designed for similar conditions, *i.e.*, for a total friction head of 2 ft and a temperature drop through the radiators of 10 deg when the maximum quantity of heat is being delivered to the building, a table such as Table 2 may be prepared from the data of Fig. 4. Having this table, the pipe sizes for the system of Example 1 can be easily selected. For example, for Sections BC and JK, each supplying 54 Mbh, the equivalent pipe length of the system is 227 ft. In the table the length shown nearest to this length is 200 ft. In the 200-ft column, a 1½-in. pipe is slightly too small and a 2-in. pipe is too large. The 1½-in. pipe will therefore be selected. For Sections CD and IJ, supplying 42 Mbh, a 1¼-in. pipe is too small and a 1½-in. pipe is too large, so 1¼ in. will be selected for the flow and 1½ in. for the return line. For larger

Table 2. Capacities of Pipes in *Mbh* (1000 Btu per Hour) and **Velocities of Water in Pipes in Inches per Second** for Forced Circulation Systems with a Total Friction Head of 2 ft and for a Maximum Temperature Drop of 10 deg[a]

1	2	3	4	5	6	7	8	9
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit						
		100	150	200	250	300	350	400
		Unit Friction Head, in Milinches						
		240	160	120	96	80	69	60
½	1	*6.2* **15**	*4.8* **12**	*4.1* **10**	*3.4* **9**	*2.9* **8**	*2.6* **7.5**	*2.4* **7**
¾	2	*13.2* **18**	*10.3* **14**	*8.6* **12**	*7.3* **11**	*6.2* **10**	*6.0* **9**	*5.5* **8.5**
1	2.3	*25.0* **22**	*19.2* **17**	*16.3* **15**	*14.4* **13**	*12.5* **12**	*12.0* **11**	*11.1* **10.5**
1¼	3.0	*52.8* **27**	*40.8* **21**	*34.8* **18**	*31.2* **16**	*27.8* **15**	*26.4* **14**	*24.0* **13**
1½	3.5	*79.2* **30**	*60.7* **23**	*51.2* **20**	*45.6* **18**	*40.8* **16**	*40.0* **15**	*36.0* **14**
2	4.0	*153.8* **36**	*120.0* **28**	*104.0* **24**	*93.5* **22**	*86.4* **20**	*81.5* **18**	*73.8* **17**
2½	6.0	*250.0* **41**	*192.0* **32**	*164.5* **28**	*149.0* **25**	*139.2* **22**	*135.8* **21**	*122.5* **19**
3	6.5	*444.0* **48**	*348.0* **37**	*294.0* **32**	*270.0* **29**	*254.0* **26**	*240.0* **24**	*223.0* **22**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 deg, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

systems, it will be economical to operate with higher friction heads, and tables may be prepared similar to Tables 3 and 4, which are based on total friction heads of 6 and 18 ft, respectively.

Example 2. Design a direct return two-pipe forced circulation system for the layout shown in Fig. 5. For this system the length of the pipe line from the boiler to the highest radiator on the farthest riser and back to the boiler is about 250 ft. There are about 16 elbow equivalents having an equivalent pipe length of about 50 ft, so the total equivalent pipe length is about 300 ft.

Solution. The same pipe size tables may be used as those developed for the reversed return system of Fig. 3. Since this system is somewhat larger than that shown in Fig. 3, Table 3 which provides for a friction head of 6 ft may be used instead of Table 2 which provides for a friction head of only 2 ft.

Referring to the column for an equivalent total length of 300 ft for Sections *AB* and *KA*, each supplying 117.6 Mbh, it will be found that a 1½-in. pipe is too small and a 2-in. pipe is too large. Consequently, a 1½-in. pipe is selected for the flow line *AB*, and a 2-in. pipe for the return line *KA*. For Sections *BC* and *JK*, each supplying 88 Mbh, a 1½-in. pipe is only slightly too small and it is selected. The remaining pipe sizes are selected in a similar manner and recorded in Fig. 5. For a temperature drop of 10 deg, 24.5 gpm of water must be circulated. The pump to select is one which has its highest efficiency when it is delivering 24.5 gpm against a 6-ft head.

TABLE 3. CAPACITIES OF PIPES IN *Mbh* (1000 BTU PER HOUR) AND **Velocities of Water in Pipes in Inches per Second** FOR FORCED CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 6 FT AND FOR A MAXIMUM TEMPERATURE DROP OF 10 DEG[a]

1	2	3	4	5	6	7	8
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit					
		200	300	400	600	800	1000
		Unit Friction Head, in Milinches					
		360	240	180	120	90	72
½	1	*7.4* **18**	*6.0* **15**	*5.0* **13**	*3.8* **10**	*3.4* **9**	*3.1* **7.5**
¾	2	*15.8* **22**	*12.7* **18**	*10.8* **16**	*8.4* **12**	*7.7* **11**	*6.7* **9**
1	2.5	*30.0* **27**	*24.0* **22**	*20.4* **19**	*15.8* **15**	*13.9* **13**	*12.5* **11**
1¼	3.3	*64.8* **33**	*52.5* **26**	*44.4* **23**	*33.6* **18**	*30.0* **16**	*26.8* **14**
1½	4.0	*96.0* **37**	*76.8* **31**	*64.8* **26**	*50.1* **20**	*44.7* **18**	*40.8* **15**
2	5.0	*192.0* **44**	*153.0* **36**	*130.0* **30**	*100.1* **24**	*90.0* **21**	*78.0* **18**
2½	6.0	*300.0* **50**	*244.0* **41**	*206.0* **35**	*161.0* **26**	*144.0* **24**	*130.0* **.21**
3	7.5	*550.0* **58**	*436.0* **48**	*368.0* **42**	*287.0* **32**	*249.0* **27**	*228.0* **24**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 deg, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

To secure a correct distribution of hot water among the several risers it is necessary, as previously stated, to introduce special resistances to balance the several risers, as follows:

The first riser is 80 ft nearer the boiler than the fifth riser. In order that the two may be balanced, *i.e.*, that they may operate under equal pressure heads, resistance must be added to the first riser equal to the friction head in the 80 ft of flow main from *B* to *F* plus that in the 80 ft of return main from *G* to *K*.

It will be noted from Table 3 that the unit friction head is about 240 milinches per foot. The total friction head in the flow and return mains between the first and fifth risers is therefore 160 × 240 or 38,400 milinches, or a little more than 3 ft, which must be supplied by additional resistance in the first riser to prevent its having an advantage over the fifth riser.

This resistance can be supplied by a calibrated and adjusted modulating valve or by an orifice resistor in a union. If the orifice resistor is to be used, its size may be selected from Table 5 as follows:

The lower part of the first flow riser supplies 28.8 Mbh. According to Table 3, it should be a 1-in. pipe and would have a velocity of 22 in. per second, if it were supplying 24 Mbh. Since it is supplying 28.8 Mbh, the velocity will be about 26 in. per second. From Table 5 it will be found that for a 1-in. pipe and a velocity of 24 in. per second, an 0.45-in. orifice will produce a loss of head of 37,000 milinches. For a velocity of 26 in. per second, the loss of head will be somewhat more, probably about 43,000 milinches; the

Table 4. Capacities of Pipes in *Mbh* (1000 Btu per Hour) and **Velocities of Water in Pipes in Inches per Second** for Forced Circulation Systems with a Total Friction Head of 18 ft and for a Maximum Temperature Drop of 10 deg[a]

1	2	3	4	5	6	7
Pipe Size (Inches)	Equivalent Length of Pipe (Feet[b])	Equivalent Total Length of Pipe in Feet in Longest Circuit				
		200	400	600	800	1000
		Unit Friction Head, in Milinches				
		1080	540	360	270	216
½	1.0	*12.7* **32**	*8.6* **23**	*7.2* **18**	*6.2* **15**	*5.5* **13**
¾	2.0	*27.5* **40**	*18.7* **28**	*15.1* **22**	*13.7* **19**	*11.5* **17**
1	2.5	*55.0* **48**	*36.8* **34**	*30.0* **27**	*26.4* **23**	*22.6* **20**
1¼	3.0	*122.0* **59**	*81.5* **42**	*66.0* **33**	*58.3* **28**	*50.5* **25**
1½	4.0	*182.0* **66**	*122.0* **46**	*98.2* **37**	*86.2* **31**	*74.2* **27**
2	5.0	*371.0* **80**	*252.0* **56**	*201.0* **45**	*180.0* **38**	*151.0* **33**
2½	7.0	*598.0* **91**	*407.0* **65**	*323.0* **51**	*287.0* **43**	*240.0* **38**
3	9.0	*1110.0* **107**	*790.0* **76**	*598.0* **60**	*527.0* **51**	*443.0* **44**

[a]For other temperature drops the capacities of pipes are to be changed correspondingly. For example, for a temperature drop of 30 deg, the capacities shown in this table are to be multiplied by 3. The velocities remain unchanged.

[b]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

difference between it and the required resistance will be about 10 per cent which is permissible, and the 0.45-in. orifice is selected.

The sizes of the orifice resistors for the second, third, and fourth risers are selected in a similar manner and found to be 0.45 in., 0.50 in., and 0.55 in., respectively.

If the design of the system of Fig. 5 is to be extremely refined, the gravity pressure heads produced by the risers should be taken into consideration. With water at 220 F and 210 F, respectively, in the risers, the gravity head is 50 milinches per foot of water column or 25 milinches per foot of flow and return pipe. The pump pressure head in this case is 240 milinches per foot of pipe, and the gravity head, being only one tenth as large as the pump head, may be neglected without serious error. This is generally done.

Temperatures of 220 F and 210 F would be used only during the coldest

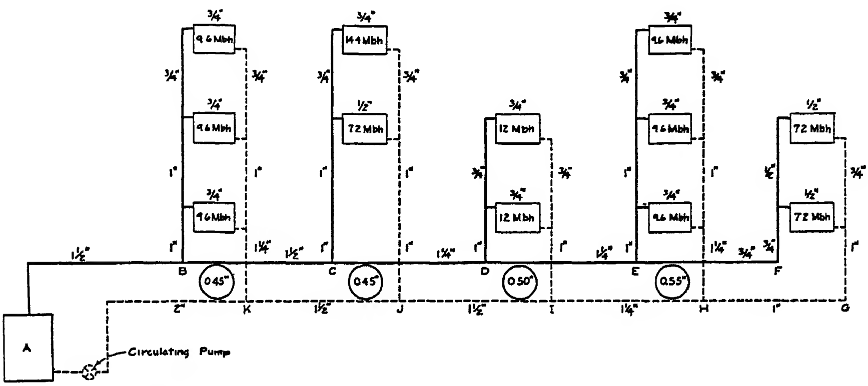


FIG. 5. A FORCED CIRCULATION DIRECT RETURN SYSTEM

weather for which the system is designed. At other times the temperatures would be lower, the temperature drop smaller, and the gravity heads smaller. The pump pressure head remains constant throughout the season if the pump is operated at a constant speed and, consequently, the gravity head is generally less than one-tenth of the pump head.

Effect of Variations in Pipe Sizes

The pipe sizes for the several parts of the system selected from the tables are only approximately correct but the resulting error should be negligible as may be seen from the following study. Assume, as an extreme case, that the error in pipe size is so large that the water flows twice as fast through one of the radiators as through the others. This would make the friction head through this radiator almost four times as large as those through the other radiators. The result would be that the water, in flowing through the radiator, would cool 5 deg instead of 10 deg. The mean water temperature in the radiator would then be 217½ F instead of 215 F, and the mean temperature difference, water to air, would be 147½ deg instead of 145 deg. The heat dissipated by the radiator would therefore be about 2 per cent more than calculated. It is evident that this difference in heat dissipation is smaller than the difference

TABLE 5. FRICTION HEADS (IN MILINCHES) OF CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN UNIONS

DIAMETER OF ORIFICES (INCHES)	VELOCITY OF WATER IN PIPE IN INCHES PER SECOND									
	2	3	4	6	8	10	12	18	24	36
					¾-in. Pipe					
0.25	1300	2900	5000	11,300	20,800	32,000	45,000			
0.30	650	1450	2500	5700	10,400	16,000	23,000	57,000		
0.35	330	740	1300	2900	5200	8000	12,000	26,000	47,000	
0.40	170	380	660	1500	2600	4000	6800	13,000	24,000	53,000
0.45		185	330	740	1300	2000	2900	6500	12,000	27,000
0.50			155	350	620	970	1400	3200	5700	13,000
0.55			75	170	300	480	700	1600	2800	6400
					1-in. Pipe					
0.35	900	2000	3500	7800	14,000	22,000	32,000			
0.40	460	1000	1800	4000	7200	12,000	17,000	37,000	65,000	
0.45	270	570	1000	2300	4100	6400	9300	21,000	37,000	
0.50	160	330	580	1400	2300	3700	5400	12,000	22,000	50,000
0.55		190	330	750	1300	2200	3000	7000	13,000	28,000
0.60			200	440	800	1300	1800	4200	7400	17,000
0.65			120	260	460	720	1100	2400	4300	10,000
					1¼-in. Pipe					
0.45	1000	2250	4000	8900	16,000	25,000	36,000			
0.50	660	1450	2600	5800	10,400	16,400	23,000	53,000		
0.55	430	950	1700	3800	6800	10,500	15,000	34,000	60,000	
0.60	280	630	1100	2500	4400	6900	10,000	22,000	40,000	
0.65	190	420	750	1700	3000	4700	6700	15,000	27,000	60,000
0.70		285	510	1150	2000	3100	4500	10,000	18,000	40,000
0.75		190	330	750	1300	2100	3000	6700	12,000	26,000
					1½-in. Pipe					
0.55	850	1900	3300	7400	13,000	21,000	30,000			
0.60	600	1300	2300	5400	8600	16,800	21,000	50,000		
0.65	400	850	1500	3600	7200	10,400	14,000	30,000	53,000	
0.70	260	600	1100	2600	4400	7000	10,000	21,000	39,000	
0.75	180	400	760	1800	3000	5000	7000	14,000	28,000	
0.80		300	540	1200	2200	3200	5000	10,200	19,000	45,000
0.85		200	380	860	1600	2300	3000	7800	13,000	30,000
					2-in. Pipe					
0.70	890	1850	3500	7400	14,000	22,300	33,000			
0.80	470	975	1800	3900	7400	11,700	17,000	37,000		
0.90	255	560	1000	2200	4200	6500	9500	20,500	38,000	
1.00	160	340	610	1320	2520	4000	5800	12,500	23,000	49,000
1.10		214	375	850	1600	2500	3700	7900	14,000	30,000
1.20			195	460	950	1360	1910	4200	8100	16,800
1.30				275	525	980	1375	3100	4400	8850

Note.—The losses of head for the orifices in the 1½-in. and 2-in. pipe were calculated from those in the smaller pipes, the calculations being based on the assumption that, for any given velocity, the loss of head is a function of the ratio of the diameter of the pipe to that of the orifice. This had been found to be practically true in the tests to determine the losses of head in orifices in ¾-in., 1-in., and 1¼-in. pipe, conducted by the Texas Engineering Experiment Station, and also in the tests to determine the losses of head in orifices in 4-in., 6-in., and 12-in. pipe, conducted by the Engineering Experiment Station of the University of Illinois, (Bulletin 109, Table 6, p. 38, Davis and Jordan).

between the calculated heat losses and the actual heat losses, and also smaller than the average difference between the calculated radiator sizes and the nearest stock sizes selected.

GRAVITY CIRCULATION

For gravity circulation, the one-pipe system shown in Fig. 6 and the two-pipe direct return system shown in Fig. 7 are probably in most common use.

The one-pipe system has the disadvantage that the radiator nearest the

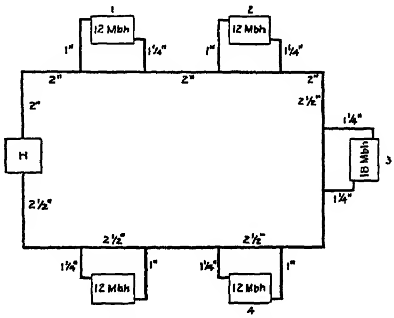


FIG. 6. A ONE-PIPE GRAVITY CIRCULATION SYSTEM

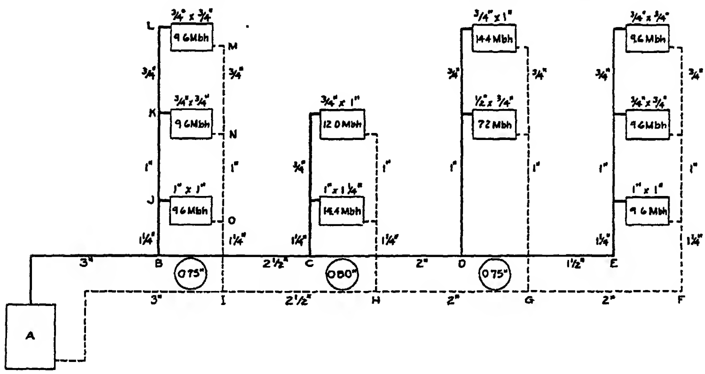


FIG. 7. A TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEM

boiler is the only one which receives water at approximately the temperature at which it leaves the boiler. All other radiators receive cooler water and must be proportionally increased in size, so the total heating surface in the system is considerably larger than that in a corresponding two-pipe system.

The pipe sizes in gravity circulation systems may be varied. As the pipe sizes are decreased, the temperature drop through the radiators, which produces circulation, is increased and it becomes necessary to increase the temperature of the water leaving the boiler so that the mean temperature in the radiator remains constant. For example, Fig. 8 shows

diagrammatically an elementary heating system which will function with either 1½-in. or 1-in. pipe. The radiator is required to deliver 27 Mbh, and the circuit consists of 30 ft of pipe and 20 elbow equivalents.

If 1½-in. pipe is used, the system will operate correctly if the water temperatures in the flow and return risers are 200 F and 180 F, respectively. The mean water temperature in the radiators will then be 190 F and, if the radiator is located in air having a temperature of 70 F, the size of the radiator must be sufficient to deliver 27 Mbh under these conditions.

If 1-in. pipe is used, the system will function correctly with water temperatures in the flow and return risers of 210 F and 170 F, or of 200 F and 160 F. In the first case, the mean water temperature is again 190 F and the same size radiator may be used as with the 1½-in. pipe, but the temperature of the water leaving the boiler must be raised from 200 F to 210 F. In the second case, the temperature of the water leaving the boiler is the same as for the 1½-in. pipe, but the mean water temperature

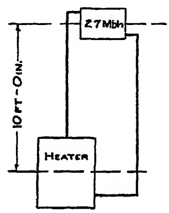


FIG. 8. AN ELEMENTARY SYSTEM

in the radiator is lowered from 190 F to 180 F, and theoretically the size of the radiator should be increased about 12½ per cent to deliver the required 27 Mbh (See Table 3, Chapter 6, 1933 GUIDE).

This indicates the extent to which pipe sizes and radiator sizes may be decreased by increasing the temperatures of the water in the boiler, as is possible in closed systems and in open systems in which the open expansion tank is located sufficiently high to secure a pressure in the boiler equal to that existing in the boiler of the closed system.

Example 3. Design a one-pipe gravity circulation system for the layout shown in Fig. 6. Assume that the main circuit consists of 150 ft of pipe, 7 elbows, and one boiler.

Solution. Replace the boiler by 3 elbow equivalents and assume that the size of the main will be about 2 in. According to Table 6, Column 2, a 2-in. elbow is equivalent to 4 ft of pipe, and the total equivalent length of the main will be about 150 plus 40, or 190 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main and that the temperature drop in the system is to be 35 deg, Table 6 may be used to determine the size of the mains. Note from Column 8, for a 200-ft length, that a 2-in. main will supply 48 Mbh and a 2½-in. main, 75.4 Mbh. Since the system to be designed is to supply 66 Mbh, a 2-in. pipe is too small and a 2½-in. pipe too large. The solution is to use some 2-in. and some 2½-in. pipe. Since the 2½-in. is nearer the correct size than the 2-in., select 2-in. pipe for the first 50 or 60 ft out of the boiler and 2½-in. for the remaining pipe back to the boiler.

Tables 7 and 8 may be used to design the radiator risers and connections. According to Table 7, for 12 Mbh the flow riser should be ¾ in. and the return riser 1 in., and the riser branches should be 1 in. and 1¼ in., respectively. Note that according to Table 8, both radiator tappings should be 1 in. To simplify the construction, select 1-in. flow risers with 1-in. riser branches and 1-in. radiator tappings. Also select 1¼-in. return risers with 1¼-in. riser branches, and 1¼-in. radiator tappings. Similarly, for 18 Mbh, select 1¼-in. flow and return risers and riser branches, and 1¼-in. radiator tappings.

TABLE 6. CAPACITIES OF MAINS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TOTAL FRICTION HEAD OF 0.6 IN., A TEMPERATURE DROP OF 35 DEG, WHEN THE MAINS ARE 4 FT ABOVE THE CENTER OF THE BOILER

	2	3	4	5	6	7	8	9	10	11
		EQUIVALENT TOTAL LENGTH OF PIPE IN FEET IN LONGEST CIRCUIT								
PIPE SIZE (INCHES)	EQUIVALENT LENGTH OF PIPE (FEET[a])	75	100	125	150	175	200	250	300	350
		UNIT FRICTION HEAD, IN MILINCHES								
		8.0	6.0	4.8	4.0	3.4	3.0	2.4	2.0	1.7
1½	3.0	*43.0*	*37.5*	*33.0*	*30.0*	*27.0*	*25.0*	*22.2*	*20.2*	*18.7*
2	4.0	*83.0*	*72.0*	*63.0*	*57.0*	*51.0*	*48.0*	*42.0*	*38.0*	*35.0*
2½	4.5	*140.0*	*115.0*	*100.0*	*90.0*	*81.5*	*75.4*	*67.2*	*61.0*	*56.0*
3	5.0	*234.0*	*204.0*	*175.5*	*160.0*	*143.0*	*133.0*	*110.0*	*107.5*	*100.0*
3½	5.5	*347.0*	*300.0*	*260.0*	*236.0*	*214.0*	*200.0*	*177.0*	*160.0*	*146.0*
4	6.0	*490.0*	*422.0*	*370.0*	*334.0*	*297.0*	*278.0*	*248.0*	*223.0*	*205.0*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

To develop a rule for determining radiator sizes, assume a system similar to that of Fig. 6, in which the total temperature drop is to be 35 deg and which is equipped with 7 radiators, all radiators dissipating equal quantities of heat. The mean temperature of the water in the radiators will be reduced 5 deg for each successive radiator. If the mean temperature of the water in the first radiator is 200 F, the mean tem-

TABLE 7. MAXIMUM CAPACITIES OF RISERS[a] IN *Mbh*, AND **Velocities of Water in Pipes in Inches Per Second** FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A DROP OF 35 DEG THROUGH EACH RADIATOR

PIPE SIZE (INCHES)		EQUIVALENT LENGTH OF PIPE (FEET[c])	1ST FLOOR[b]			2ND FLOOR	3RD AND 4TH FLOORS
				Vel. (Ft per Sec.)[d]			
Flow	Return		*Mbh*	Flow	Return	*Mbh*	*Mbh*
½	½	1.0				*5*	*6.2*
½	¾					*6.4*	*8.0*
¾	¾	1.5	*9*	2.3	2.3	*10.1*	*14.0*
¾	1		*12*	3.2	2.0	*12.8*	*17.1*
1	1	2.0	*18*	2.5	2.5	*20*	*26.0*
1	1¼		*21*	3.0	2.0	*25.2*	*34*
1¼	1¼	3.0	*26*	3.0	3.0	*43*	*55*
1¼	1½		*34*	4.0	2.5		
1½	1½	3.5	*48*	3.0	3.0		

[a]This table is based on pressure heads of 450, 1800, 3150, and 4500, respectively, for the first, second, third, and fourth floor radiators, and on friction heads of 200 milinches for the first floor radiators and connections, and 700 milinches for all other radiators and their connections.

[b]The riser branches, the piping which connects the risers to the mains, are to be one size larger than the risers.

[c]Approximate length of pipes in feet equivalent to one elbow in friction head. This value varies with the velocity.

[d]Velocities apply to the riser branches.

perature of the water in the seventh radiator will be 170 F, and, according to Table 3, Chapter 6, of the 1933 GUIDE, the heat dissipation of these two radiators will be to each other as 868 is to 617, or as 140 is to 100, and therefore if the last radiator is to dissipate as much heat as the first, its size must be 40 per cent larger.

Example 4. Design a two-pipe, direct return, gravity circulation system for the layout shown in Fig. 7. Assume that the main circuit from the boiler to the farthest flow riser and from the farthest return riser back to the boiler consists of 160 ft of pipe, 6 elbows, and 1 boiler.

Solution. Replacing the boiler by 3 elbow equivalents and assuming that the largest size of the main will be about 3 in., the total equivalent length of the main will be 160 plus 45, or 205 ft. Assuming that the center of the boiler will be about 4 ft lower than the horizontal portion of the main, and that the temperature drop will be 35 deg for the system, the pressure head caused by the difference in weight between the water in the

TABLE 8. MAXIMUM CAPACITIES OF RADIATOR CONNECTIONS IN *Mbh*, FOR ONE-PIPE AND FOR TWO-PIPE DIRECT RETURN GRAVITY CIRCULATION SYSTEMS WITH A TEMPERATURE DROP OF 35 DEG THROUGH EACH RADIATOR

PIPE SIZE		EQUIVALENT LENGTH OF PIPE (FEET[a])	1ST FLOOR	2ND, 3RD, AND 4TH FLOORS
Flow	Return		*Mbh*	*Mbh*
½	½	1.0	*4.1*	*5.9*
½	¾		*5.2*	*7.5*
¾	¾	1.5	*7.0*	*10.5*
¾	1		*9.1*	*13.0*
1	1	2.0	*12.5*	*17.8*
1	1¼		*17.5*	*23.2*
1¼	1¼	3.0	*23.3*	*33.2*

[a]Approximate length of pipe in feet equivalent to one elbow in friction head. This value varies with the velocity.

flow and return risers joining the mains to the boiler will be about 0.6 in. of water, or about one-fortieth of the pressure head produced by the circulating pump selected for the system of Fig. 3.

Table 6 may be used to determine the size of the main as follows: Refer to Column 8 and note that for Sections *AB* and *IA*, which supply 105.6 Mbh, a 3-in. pipe is too large and a 2½-in. pipe is too small; hence, select 2½ in. for Section *AB* and 3 in. for Section *IA*. For Sections *BC* and *HI*, which supply 76.8 Mbh, a 2½-in. pipe is almost exactly the correct size and is selected for both sections.

For the forced circulation system of Fig. 5, the pressure head produced by the circulating pump is used to force the water through.the mains and also through the risers. Gravity circulation systems have two distinct pressure heads. One is produced by the difference in weight of the water in the flow and return risers adjacent to the boiler, and is the boiler pressure head, which in this case is 0.6 in. The other pressure head is produced by the difference in weight of the water in the flow and return risers adjacent to the radiators, and is the radiator pressure head. If the temperature drop through the radiators is about 35 deg, and if the story heights of the building are 9 ft and the distance from the center of the first floor radiator to the average level of the main is 3 ft, the radiator pressure head of the first floor radiator is about 450 milinches and the pressure heads of the radiators on the upper floor are 1350 milinches greater than those on the next lower floors.

Tables 6 and 7 are based on the assumption that the boiler pressure head must be equal to the friction head in the mains, and that the several radiator pressure heads must be equal to the respective radiator and riser friction heads.

To design the radiator risers, use Table 7 and begin with the set nearest the boiler. The first floor risers must supply 28.8 Mbh. According to the table, 1¼-in. flow and return risers will supply 26.0 Mbh; if the return riser is increased to 1½ in., the capacity will be increased to 34.0 Mbh. This is considerably larger than necessary, and 1¼-in. flow and return risers are selected. However, it must be remembered that the riser

branches, which are the connections from the flow and return mains to the flow and return risers, are to be one size larger than the risers.

The second floor risers must supply 19.2 Mbh. According to the table, the capacity of 1-in. flow and return risers is 20.0 Mbh, and that size is selected.

The third floor risers must supply 9.6 Mbh. If a ½-in. flow and a ¾-in. return riser are used, the capacity will be 8.0 Mbh; if both risers are ¾ in., the capacity will be 14.0 Mbh. The ¾-in. pipe is selected for both risers.

To design the radiator connections, use Table 8 and note that for the first floor radiator connections the capacity of a ¾-in. flow and 1-in. return is 9.1 Mbh, and that of a 1-in. flow and a 1-in. return is 12.5 Mbh. The former is more nearly the correct size, but since it is difficult to secure a good flow through first floor radiators, the 1-in. flow and return connection is selected. For the two upper floors, the capacity of a ¾-in. flow and return connection is 10.5 Mbh, and that size is used.

As explained in the design of the forced circulation system of Fig. 5, the two-pipe direct return system of Fig. 7 will not function correctly unless its four sets of risers are balanced among themselves. This necessary balancing is accomplished by adding resistances to all risers, except the one farthest from the boiler, equal to the excess boiler pressure heads available for those risers above the boiler pressure head available for the farthest riser. For example, the first set of risers is 60 ft nearer the boiler than the last set. Since the flow and return mains are designed for a friction head of 3 milinches per foot (See Table 6, Column 8), the boiler pressure head available for the first set of risers is 360 milinches in excess of that available for the fourth set. The velocity in the riser branch is 3 in. per second (See Table 7) and, therefore, according to Table 5, an 0.75-in. orifice in a 1½-in. union should be used. This will provide a resistance of about 400 milinches. In the same manner it is found that for the second set of risers a resistance of 240 milinches is required and that an 0.80-in. orifice in a 1½-in. union will provide a resistance of 300 milinches. For the third set of risers, a resistance of 120 milinches is required and an 0.75-in. orifice in a 1¼-in. union will provide sufficient resistance.

MECHANICAL CIRCULATION

Circulating pumps for hot water systems may be used to provide the motive head for forced circulation systems as already described, or to improve the operation of gravity-designed systems. Small specially-designed centrifugal pumps installed on a by-pass with the necessary gate or check valves near the point where the return main enters the heater may be employed. Specially-designed, electrically-driven, propeller-type circulating pumps or units may also be employed. The latter are usually installed directly in the return main and are available for all commercial pipe sizes used for hot water heating. The motor switch may be under manual control, automatic control using thermostatic elements, or tied in with the oil or gas burner switch which starts and stops the burner. For large capacities these units may be installed in multiple.

For exceptionally large installations such as central heating plants, circulating pumps of the centrifugal single stage type, having an average operating efficiency of 70 per cent against heads up to 125 ft, are sometimes used. It is generally advisable to install the pumps in duplicate to provide for contingencies and to insure continuous operation. In such cases each pump may be made equal to two-thirds of the maximum capacity required.

EXPANSION TANKS

When water at ordinary temperatures is heated or cooled, its volume is increased or decreased. This variation in the volume of the water in a heating system is generally provided for by means of an expansion tank into which the water can flow from the system during the heating-up periods and from which it can flow back into the system during the cooling-down periods.

The expansion tank may be open or closed. In an open expansion tank (Fig. 9), the water is subjected to atmospheric pressure and can expand freely without a material increase in pressure. In a closed expansion tank (Fig. 10), the water is subjected to the pressure of the compressed air

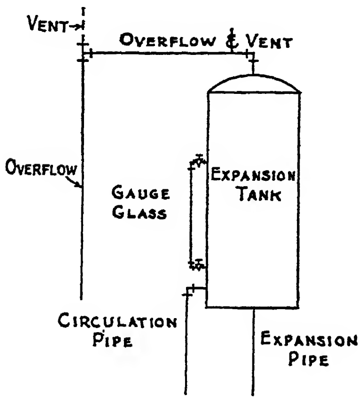


FIG. 9. AN OPEN EXPANSION TANK

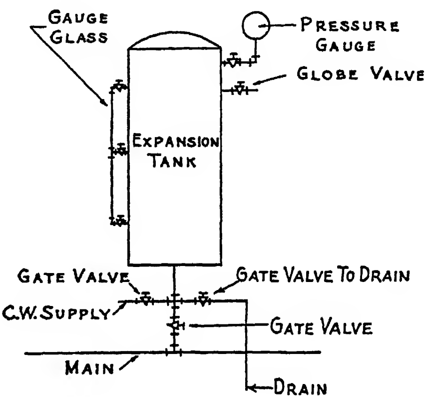


FIG. 10. A CLOSED EXPANSION TANK

within the tank, and as the water expands, the volume of the air in the tank is decreased and its pressure increased.

The open expansion tank must be placed at a sufficient elevation above the highest radiator to prevent boiling when the water in that radiator is at the highest temperature to which it is to be heated. For example, if the water is to be heated to 225 F on extremely cold days, the absolute pressure on the water in the highest radiator must be at least 19 lb per square inch. This pressure will be secured if the open expansion tank is located 15 ft above the highest radiator. If a closed expansion tank is used and is located 30 ft below the highest radiator, an absolute pressure of about 32 lb per square inch must be maintained in the expansion tank if the water in the highest radiator is to be heated to 225 F without danger of boiling.

The type of expansion tank used in a heating system, whether open or closed, has no influence on the operation of the system. The only function performed by the expansion tank is to provide for the variation in the volume of the water in the system, and at the same time to maintain a sufficient pressure in the system to prevent boiling when the water is at the highest temperature for which the system is designed. The use of an expansion tank may be dispensed with when the heating system is allowed to *float* on the water system, *i.e.*, when the connection between

the heating system and the water system is kept open so that the water system replaces the expansion tank.

The capacity of the expansion tank should be at least twice the increase in volume produced when the water in the system is heated from its normal to its maximum temperature. When 25 gal of water are heated from 40 F to 200 F, the volume of water increases to 26 gal. A safe rule, therefore, is to make the water capacity of the expansion tank equal to 10 per cent of the capacity of the heating system.

In a forced circulation system, the expansion tank should be connected to the return main near the circulating pump. In a gravity circulation system, the expansion tank should be connected to the flow riser so that air liberated from the water in the boiler may escape through the expansion tank, except where it is desired to maintain a temperature higher than 212 F, in which case the connection should be in the return main to prevent possible boiling in the expansion tank.

The expansion tank should be protected so that the water in the tank or in the connecting pipe lines cannot freeze. If the water should freeze and the water in the system be heated to cause further expansion, the resulting force will burst the boiler or some other portion of the system.

INSTALLATION DETAILS

The detailed installation of the pipe system should be governed by four fundamental rules:

1. All piping must be pitched either up or down so that all gases which are liberated from the water can move freely to a vented section of the system. Whenever practicable, the pipe line should be pitched so that gases flowing to a vent will flow in the same direction as the water. When a pipe system cannot be installed without creating *air pockets*, that is, sections in the system from which liberated gases cannot escape, such sections must be provided with automatic air relief valves or with air valves which may be operated manually when necessary.

2. All piping must be arranged so that the entire system can be drained, either to permit alterations or repairs, or to prevent freezing if the system is not to be operated during a cold period.

It is well to install a gate valve and union in every riser near the main to permit the draining of individual risers without draining the entire system. It is also well, in large installations, to divide the system into branches and to provide each branch with unions and valves so that any one branch can be drained without disturbing the remaining ones.

The dividing of large heating systems into branches or zones and providing each zone with individual valves has the further advantage of permitting a varying temperature control. For example, if a building is equipped with a forced circulating system and if the south rooms are on one branch of the main and the north rooms are on a separate branch, the valves may be set so that the water will circulate through the north branch with a temperature drop of, say, 10 deg, and through the south branch with a temperature drop of, say, 20 deg, thus delivering less heat to the south rooms than to the north rooms. This arrangement is especially valuable when the regulating valves are controlled thermostatically by the temperatures in the two zones, because no matter how accurately the heating system may have been designed, the heat demand of any group of rooms varies with sunshine and with wind velocity, and these intermittent variations can be provided for only by the individual control made possible by changing the valve settings controlling the heat supplied to particular groups of rooms.

3. All piping must be installed so that it is free to expand and contract with changes of temperature without producing undue stresses in the pipes or connections. For this purpose it is generally sufficient to allow for a variation in length of 1 in. for 100 ft of pipe.

4. The pipe system must be installed so that each circuit has its correct friction head.

To bring this about, it is necessary in some cases to minimize the friction, *i.e.*, to make the pipe line as short as possible and to provide as few fittings as possible; and in other cases it is necessary to increase the length of the pipe and the number of fittings so that, for every circuit, the friction head will be equal to the available pressure head.

The connections from the boiler to the mains should be short and direct, to reduce the friction head. It is frequently possible to avoid an elbow and to reduce the length of the pipe by running the pipe in a diagonal direction, either in a horizontal or in a vertical plane.

The mains and branches should pitch up and away from the heater, generally not less than 1 in. in 10 ft. The flow main should always be covered; the return main should be covered except where it is to provide the heating surface for the basement.

The connections from mains to branches and to risers should be such that circulation through the risers will start in the right direction. Hence, in a one-pipe system the flow connection must be nearer the heater than the return connection. In a correctly-designed two-pipe system, the pressure in the flow main is higher than that in the return main, and a slight variation in the distances of the flow and return connections from the heater is not material; but it is generally best to have the two connections about equally distant from the heater.

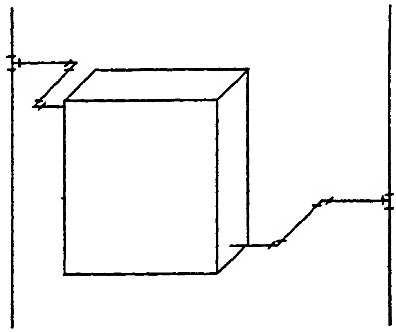


Fig. 11. Method of Connecting Radiator to Allow for Expansion of Pipe

In some cases it may be advisable to take the flow connection off the top of the main and the return connection from the side, but in most cases both connections should be at an angle of 45 deg. This method shortens the lines and substitutes 45-deg ells for 90-deg ells.

Preferably, connection of the flow riser to a radiator should be to the upper tapping, and connection of the return riser to a radiator should be to the lower tapping. When hot water enters at the top of a radiator it will distribute itself along the entire length of the radiator, and as it cools it will settle gradually to the bottom; the cool water may then be taken out of the radiator at either end.

With forced circulation and high velocities, it is advisable to let the water enter at the top of the radiator and leave at the bottom of the opposite end. With gravity circulation and low velocities it makes little difference whether the water leaves at the end at which it enters or at the opposite end.

The connections of the risers to the radiators should be such that provision is made for the vertical expansion of the risers. This can be accomplished as indicated in Fig. 11 by using one tee and two ells for each connection. These connections should be pitched upward or downward, whichever may be necessary to prevent the formation of air pockets and to permit draining.

Chapter 34

PIPE, FITTINGS, WELDING

Designation of Pipe, Types of Pipe, Expansion and Contraction, Fittings, Valves, Corrosion, Pipe Welding

PIPE used for heating and ventilating installations is made either by shaping sheets of metal into cylindrical form and welding the edges together, or by forming or drawing from a solid billet. In the latter case, it is termed *seamless tubing* or *seamless pipe*. Welded pipe usually is made by either the forge lap-weld or butt-weld process, depending upon the size.

DESIGNATION OF PIPE

Wrought pipe up to 12 in. in diameter is usually designated by its nominal internal diameter which is slightly different from its actual internal diameter, being considerably less in the smaller sizes than the actual dimension. There are three weights of wrought iron and steel pipe commonly used, known as *standard, extra strong,* and *double-extra strong.* Because of the necessity of maintaining the same external diameter in all three weights, for the same nominal size, the added wall thickness is obtained by decreasing the internal diameter.

The term *full weight,* when applied to sizes below 8 in., means that the pipe is up to the nominal weight per foot. When applied to sizes between 8 and 12 in., inclusive, it often indicates that the pipe has the heaviest of the various wall thicknesses listed. In sizes 14 in. and upward pipe is designated by its outside diameter (O.D.) and the wall thickness is specified. The dimensions of standard and extra strong pipe are given in Tables 1 and 2. The use of double-extra strong pipe is limited almost entirely to high pressure hydraulic work.

TYPES OF PIPE

Wrought-Steel Pipe. Because of its low price, the great bulk of wrought pipe used at the present time is of wrought steel. The material used for steel pipe is a mild steel made either by the Bessemer or basic open-hearth process or by the electric furnace.

Wrought-Iron Pipe. The correct definition of wrought iron as suggested by the *International Society for Testing Materials* is "malleable iron which is aggregated from pasty particles without subsequent fusion and contains so little carbon that it does not harden usefully when cooled rapidly."

Identification of Wrought-Iron and Steel Pipe. Wrought-iron pipe is marked at the mill with a spiral line the entire length of each bar, either knurled into the metal or painted in red or other bright color. Otherwise

TABLE 1. STANDARD WROUGHT PIPE

Table of Standard Dimensions

Size Inches	Diameters		Nominal Thickness Inches	Circumference		Transverse Areas			Length of Pipe per Square Foot of		Nominal Weight per Foot	
	External Inches	Approximate Internal Inches		External Inches	Internal Inches	External Sq In	Internal Sq In	Metal Sq In	External Surface	Internal Surface	Plain Ends	Threaded and Coupled
1/8	0.405	0.269	0.068	1.272	0.845	0.129	0.057	0.072	9.431	14.199	0.244	0.245
1/4	0.540	0.364	0.088	1.696	1.144	0.229	0.104	0.125	7.073	10.493	0.424	0.425
3/8	0.675	0.493	0.091	2.121	1.549	0.358	0.191	0.167	5.658	7.747	0.567	0.568
1/2	0.840	0.622	0.109	2.639	1.954	0.554	0.304	0.250	4.547	6.141	0.850	0.852
3/4	1.050	0.824	0.113	3.299	2.589	0.866	0.533	0.333	3.637	4.635	1.130	1.134
1	1.315	1.049	0.133	4.131	3.296	1.358	0.864	0.494	2.904	3.641	1.678	1.684
1 1/4	1.660	1.380	0.140	5.215	4.335	2.164	1.495	0.669	2.301	2.767	2.272	2.281
1 1/2	1.900	1.610	0.145	5.969	5.058	2.835	2.036	0.799	2.010	2.372	2.717	2.731
2	2.375	2.067	0.154	7.461	6.494	4.430	3.355	1.075	1.608	1.847	3.652	3.678
2 1/2	2.875	2.469	0.203	9.032	7.757	6.492	4.788	1.704	1.328	1.547	5.793	5.819
3	3.500	3.068	0.216	10.996	9.638	9.621	7.393	2.228	1.091	1.245	7.575	7.616
3 1/2	4.000	3.548	0.226	12.566	11.146	12.566	9.886	2.680	0.954	1.076	9.109	9.202
4	4.500	4.026	0.237	14.137	12.648	15.904	12.730	3.174	0.848	0.948	10.790	10.889
5	5.563	5.047	0.258	17.477	15.856	24.306	20.006	4.300	0.686	0.756	14.617	14.810
6	6.625	6.065	0.280	20.813	19.054	34.472	28.891	5.581	0.576	0.629	18.974	19.185
8	8.625	8.071	0.277	27.096	25.356	58.426	51.161	7.265	0.442	0.473	24.696	25.000
8	8.625	7.981	0.322	27.096	25.073	58.426	50.027	8.399	0.442	0.478	28.554	28.809
10	10.750	10.192	0.279	33.772	32.019	90.763	81.585	9.178	0.355	0.374	31.201	32.000
10	10.750	10.136	0.307	33.772	31.843	90.763	80.691	10.072	0.355	0.376	34.240	35.000
10	10.750	10.020	0.365	33.772	31.479	90.763	78.855	11.908	0.355	0.381	40.483	41.132
12	12.750	12.000	0.375	40.055	37.699	127.676	113.097	14.579	0.299	0.318	49.562	50.706

there is little difference in the appearance of wrought-iron and steel pipe but there are several tests which readily identify the materials. The fracture of wrought-iron pipe is ragged, dull gray, and fibrous. By hammering a piece of pipe flat the fracture can easily be observed. The fracture of steel is even, bright and crystalline. Wrought-iron pipe, when threads are cut on it, gives a crumbled chip, due to the fibrous structure. Steel pipe, when threaded, gives a long spiral chip. A microscopic examination of polished and etched specimens is an infallible test. The presence of slag in the wrought iron is unmistakable, while the steel is almost without slag.

Seamless Pipe. Steel pipe or *tubing* without the lap or butt weld is frequently used for high-pressure work. Its advantages are its somewhat greater strength, permitting the use of a thinner wall, and in the small sizes its freedom from the tendency of welded pipe to split at the weld occasionally when bent. Furthermore, the inherent physical properties of the steel used in its manufacture are somewhat better, and a variety of combinations of carbon content and heat treatment are available.

Seamless pipe is made by either of two different processes, depending upon the size. The piercing process is used in the smaller sizes and the cupping process is sometimes used for tubing of 6 to 8 in. diameter.

Cast Ferrous Pipe. There are now available several types of cast ferrous metal pipe made of a good grade of cast-iron with additions of nickel, chromium, etc. This pipe is available in sizes from 1½ in. to 6 in. and standard lengths of 5 or 6 ft with external and internal diameters closely approximating those of extra strong wrought pipe. Cast ferrous pipe may be had coupled, bevelled for welding, or with ends plain or grooved for the several types of couplings. It is readily cut, and threaded, as well as welded. The fact that it is readily welded enables the manufacturers to supply the pipe in any lengths practical for handling.

Alloy Metal Pipe. Steel pipe bearing a small alloy of copper is somewhat more resistant to corrosion than plain steel pipe. It is recommended usually for atmospheric corrosive conditions, *i.e.*, for corrosion caused by alternate exposure to air and water.

Copper Pipe. Copper and brass pipe are used to some extent in heating work. Special fittings are also available. While the friction loss through copper pipe is slightly greater than for steel (for hot water), this is offset by the smaller loss in the fittings[1]. Therefore, for practical purposes there is little difference between the two. (See Chapter 33).

EXPANSION AND CONTRACTION

The proper provision for the expansion and contraction of piping must be made in all cases where water, steam, or gas is to be used at high temperatures, and is usually accomplished by directional changes, long sweep bends or expansion joints. In instances where it is impracticable to install the piping to obtain the required flexibility by directional changes, it may be advantageous to insert expansion bends in the line. Such expansion bends may be (1) a complete unit such as the expansion *U*-bend, the

[1]See Loss of Head in Copper Pipe and Fittings, by F. E. Giesecke and W. H. Badgett (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

TABLE 2. EXTRA STRONG WROUGHT PIPE

Table of Standard Dimensions

Size Inches	Diameters		Nominal Thickness Inches	Circumference		Transverse Area			Length of Pipe per Square Foot of		Length of Pipe Containing One Cubic Foot Feet	Nominal Weight per Foot Plain Ends Pounds
	External Inches	Approximate Internal Inches		External Inches	Internal Inches	External Sq In	Internal Sq In	Metal Sq In	External Surface	Internal Surface		
1/8	0.405	0.215	0.095	1.272	0.675	0.129	0.036	0.093	9.431	17.766	3966.392	0.314
1/4	0.540	0.302	0.119	1.696	0.949	0.229	0.072	0.157	7.073	12.648	2010.290	0.535
3/8	0.675	0.423	0.126	2.121	1.329	0.358	0.141	0.217	5.658	9.030	1024.689	0.738
1/2	0.840	0.546	0.147	2.639	1.715	0.554	0.234	0.320	4.547	6.995	615.017	1.087
3/4	1.050	0.742	0.154	3.299	2.331	0.866	0.433	0.433	3.637	5.147	333.016	1.473
1	1.315	0.957	0.179	4.131	3.007	1.358	0.719	0.639	2.904	3.991	200.193	2.171
1 1/4	1.660	1.278	0.191	5.215	4.015	2.164	1.283	0.881	2.301	2.988	112.256	2.996
1 1/2	1.900	1.500	0.200	5.969	4.712	2.835	1.767	1.068	2.010	2.546	81.487	3.631
2	2.375	1.939	0.218	7.461	6.092	4.430	2.953	1.477	1.608	1.969	48.766	5.022
2 1/2	2.875	2.323	0.276	9.032	7.298	6.492	4.238	2.254	1.328	1.644	33.976	7.661
3	3.500	2.900	0.300	10.996	9.111	9.621	6.605	3.016	1.091	1.317	21.801	10.252
3 1/2	4.000	3.364	0.318	12.566	10.568	12.566	8.888	3.678	0.954	1.135	16.202	12.505
4	4.500	3.826	0.337	14.137	12.020	15.904	11.497	4.407	0.848	0.998	12.525	14.983
5	5.563	4.813	0.375	17.477	15.120	24.306	18.194	6.112	0.686	0.793	7.915	20.778
6	6.625	5.761	0.432	20.813	18.099	34.472	26.067	8.405	0.576	0.663	5.524	28.573
8	8.625	7.625	0.500	27.096	23.955	58.426	45.663	12.763	0.442	0.500	3.154	43.388
10	10.750	9.750	0.500	33.772	30.631	90.763	74.662	16.101	0.355	0.391	1.929	54.735
12	12.750	11.750	0.500	40.055	36.914	127.676	108.434	19.242	0.299	0.325	1.328	65.415

double-offset expansion *U*-bend, or the circle bend (Fig. 1); (2) a built-up bend composed of several of the smaller curved pieces illustrated in Fig. 1; (3) a combination of straight pieces with such curved pieces or cast ells; (4) a so-called *square* bend made up entirely of straight pieces, cast ells, etc.

All risers must be anchored and safeguarded so that the difference in length when hot from the length when cold shall not disarrange the normal and orderly provisions for drainage of the branches.

It is especially necessary with light-weight radiators so to anchor and so to give freedom for expansion of the piping that no strain therefrom shall be allowed to distort the radiators. When expansion strains from the pipes are permitted to reach these light metal heaters they usually emit sounds of distress which are exceedingly troublesome.

The linear expansion of the pipe can be determined from Table 3. The elongation values in this table were computed from the following formula:

$$L_t = L_0 \left[1 + a \left(\frac{t - 32}{1000} \right) + b \left(\frac{t - 32}{1000} \right)^2 \right] \tag{1}$$

where

L_t = length at temperature t degrees Fahrenheit, feet.
L_0 = length at 32 F, feet.
t = final temperature, degrees Fahrenheit.
a and b are constants as follows:

METAL	a	b
Cast-Iron	0.005441	0.001747
Steel	0.006212	0.001623
Wrought-Iron	0.006503	0.001622
Copper	0.009278	0.001244

In selecting expansion fittings it is well to allow a margin of 25 per cent between the computed expansion and the actual allowable travel of the expansion fitting to provide for inaccuracies in assembling, etc. Table 4 gives the dimensions of expansion offsets and bends required to take care of different amounts of expansion.

FITTINGS

Fittings for joining the separate lengths of pipe together are made in a variety of forms, and are either screwed or flanged, the former being generally used for the smaller sizes of pipe up to and including 3½ in., and the latter for the larger sizes, 4 in. and above. Screwed fittings of large size as well as flanged fittings of small size are also made and are used for certain classes of work at the proper pressure.

The material used for fittings is generally cast-iron, but in addition to this malleable iron, steel and steel alloys are also used, as well as various grades of brass or bronze. The material to be used depends on the character of the service and the pressure.

As in the case of pipe, there are several weights of fittings manufactured, designed to be used with pipe of corresponding grade. These varia-

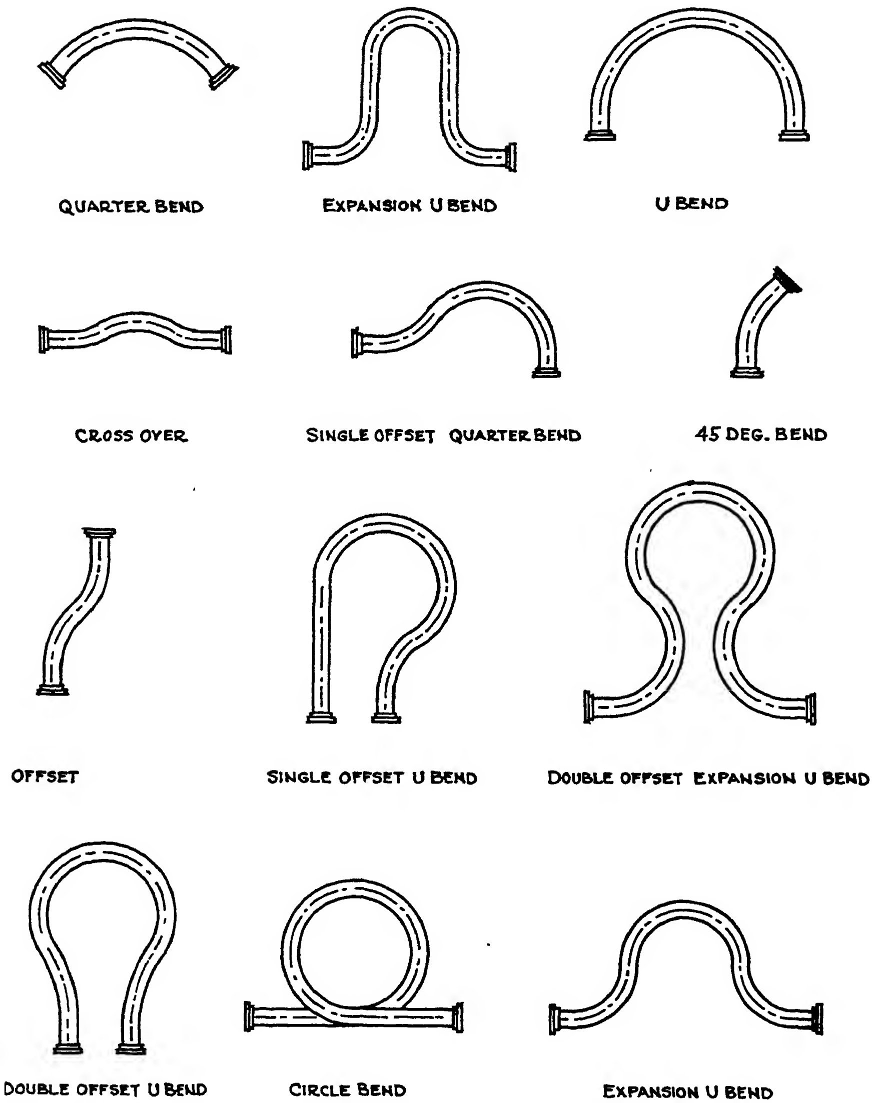


FIG. 1. COMMON TYPES OF PIPE BENDS

tions in weights are known as (1) low-pressure fittings for steam working pressures of 25 lb (cast-iron flanged only, sizes 4 in. and larger), (2) standard fittings for saturated steam working pressures of 125 lb, and (3) extra heavy fittings for saturated steam working pressures of 250 lb. These last fittings are suitable for cold water working pressures of 350 lb and are usually tested to twice the steam working pressure, or 500 lb cold hydrostatic.

Screwed fittings include: nipples or short pieces of pipe of varying

TABLE 3. THERMAL EXPANSION OF PIPE IN INCHES PER 100 FT[a]
(For superheated steam and other fluids refer to temperature column)

Saturated Steam			Elongation in Inches per 100 ft from −20 F up				Saturated Steam		Elongation in Inches per 100 ft from −20 F up			
Vacuum Inches of Hg.	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe	Pressure Pounds per Square Inch Gage	Temperature Degrees Fahrenheit	Cast-Iron Pipe	Steel Pipe	Wrought Iron Pipe	Copper Pipe
		−20	0	0	0	0	664.3	500	3.847	4.296	4.477	6.110
		0	0.127	0.145	0.152	0.204	795.3	520	4.020	4.487	4.677	6.352
		20	0.255	0.293	0.306	0.442	945.3	540	4.190	4.670	4.866	6.614
		40	0.390	0.430	0.465	0.655	1115.3	560	4.365	4.860	5.057	6.850
29.39	....	60	0.518	0.593	0.620	0.888	1308.3	580	4.541	5.051	5.268	7.123
28.89	....	80	0.649	0.725	0.780	1.100	1525.3	600	4.725	5.247	5.455	7.388
27.99	....	100	0.787	0.898	0.939	1.338	1768.3	620	4.896	5.437	5.660	7.636
26.48	....	120	0.926	1.055	1.110	1.570	2041.3	640	5.082	5.627	5.850	7.893
24.04	....	140	1.051	1.209	1.265	1.794	2346.3	660	5.260	5.831	6.067	8.153
20.27	....	160	1.200	1.368	1.427	2.008	2705	680	5.442	6.020	6.260	8.400
14.63	....	180	1.345	1.528	1.597	2.255	3080	700	5.629	6.229	6.481	8.676
6.45	....	200	1.495	1.691	1.778	2.500		720	5.808	6.425	6.673	8.912
	2.5	220	1.634	1.852	1.936	2.720		740	6.006	6.635	6.899	9.203
	10.3	240	1.780	2.020	2.110	2.960		760	6.200	6.833	7.100	9.460
	20.7	260	1.931	2.183	2.279	3.189		780	6.389	7.046	7.314	9.736
	34.5	280	2.085	2.350	2.465	3.422		800	6.587	7.250	7.508	9.992
	52.3	300	2.233	2.519	2.630	3.665		820	6.779	7.464	7.757	10.272
	74.9	320	2.395	2.690	2.800	3.900		840	6.970	7.662	7.952	10.512
	103.3	340	2.543	2.862	2.988	4.145		860	7.176	7.888	8.195	10.814
	138.3	360	2.700	3.029	3.175	4.380		880	7.375	8.098	8.400	11.175
	180.9	380	2.859	3.211	3.350	4.628		900	7.579	8.313	8.639	11.360
	232.4	400	3.008	3.375	3.521	4.870		920	7.795	8.545	8.867	11.625
	293.7	420	3.182	3.566	3.720	5.118		940	7.989	8.755	9.089	11.911
	366.1	440	3.345	3.740	3.900	5.358		960	8.200	8.975	9.300	12.180
	451.3	460	3.511	3.929	4.096	5.612		980	8.406	9.196	9.547	12.473
	550.3	480	3.683	4.100	4.280	5.855		1000	8.617	9.421	9.776	12.747

[a]From Piping Handbook, by Walker and Crocker. This table gives the expansion from −20 F to the temperature in question. To obtain the amount of expansion between any two temperatures take the difference between the figures in the table for those temperatures. For example, if a steel pipe is installed at a temperature of 60 F and is to operate at 300 F, the expansion would be 2.519 − 0.593 = 1.926 in.

lengths; couplings, usually of wrought iron only; elbows for turning angles of either 45 deg or 90 deg; return bends, which may be of either the close or open pattern, and may be cast with either a back or side outlet; tees; crosses; laterals or *Y* branches; and a variety of plugs, bushings, caps, lock-nuts, flanges and reducing fittings. Reducing fittings as well as bushings, both of which are used in changing from one pipe size to another, may have the smaller connection tapped eccentrically to permit free drainage of the water of condensation in steam lines or free escape of air in water lines.

Flanged fittings are generally used in the best practice for connecting all piping above 6 in. in diameter. While screwed fittings may be used for the larger sizes and are satisfactory under the proper working conditions, it will be found difficult either to make or to break the joints in these large sizes.

The dimensions of elbows, tees and crosses for 125 lb cast-iron screwed fittings are given in Table 5, whereas the dimensions for 125 lb cast-iron flanged fittings are given in Tables 6 and 7.

A special type of joint known as the *Sarlun* joint consists of a lip which is provided for welding to make the joint fluid tight, while mechanical strength is provided by bolted flanges. Another type of pipe joint, the *lap flange*, is made by using loose flanges on lengths of pipe whose ends are lapped to give a bearing surface for a gasket or metal joint.

TABLE 4. LENGTH OF EXPANSION OFFSETS AND BENDS FOR PROPER EXPANSION OF PIPE

TOTAL EXPANSION IN INCHES[a]	FEET OF PIPE AND OFFSET OR U-BEND FOR DIFFERENT DIAMETERS OF PIPE									
	2″	3″	4″	5″	6″	8″	10″	12″	14″	16″
1	11	13	15	17	19	21	23	25	27	30
2	15	18	21	23	26	29	32	35	38	42
3	18	22	26	29	32	36	40	43	48	52
4	21	26	30	34	37	42	47	50	56	58
5	24	30	34	38	41	47	53	57	63	65
6	27	33	37	41	45	52	58	63	69	71
7	30	36	40	44	48	56	62	68	74	----
8	32	39	43	47	52	60	66	72	----	----

[a]This column shows the total expansion the offset will take care of without a cold strain. Although some engineers allow an increase of 40 per cent in the allowable expansion if the pipe is given a suitable initial cold strain when made up, it is regarded as better practice not to allow for it.

The length of pipe in the expansion piece should be the same whether in the form of a single right-angle offset or double offset or U-bend.

The lengths of arms are figured for 12,000 lb per square inch tension for wrought iron pipe. If steel pipe is used this is good for 16,000 lb per inch so that the arm will take care of ⅓ more expansion.

VALVES

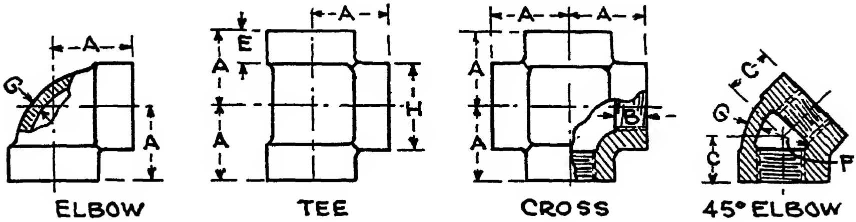
Valves are made with both threaded and flanged ends for screwed and bolted connections just as are pipe fittings.

The material used for valves of small size is generally brass or bronze for low pressures and forged steel for high pressures, while in the larger sizes either cast-iron, cast-steel or some of the steel alloys are employed. Practically all iron or steel valves intended for steam or water work are bronze-mounted or trimmed, and valves for acids, ammonia and corrosive gases are of iron throughout.

Brass, bronze, and iron valves are generally designed for standard or extra heavy service, the former being used up to 125 lb and the latter up to 250 lb saturated steam working pressure, although most manufacturers also make valves for medium pressure up to 175 lb steam working pressure. The more common types are gate valves or straightway valves, globe valves, angle valves, check valves and automatic valves, such as reducing and back-pressure valves.

A special class of valves is required for controlling the steam and hot water supply to radiators. These valves are of brass, usually of the angle type, modified to suit the service requirements, and are often arranged with graduated heads and lever handles in order to indicate the relative opening of the valve port in any position.

TABLE 5. TENTATIVE AMERICAN STANDARD DIMENSIONS OF ELBOWS, 45 DEG ELBOWS, TEES, AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON SCREWED FITTINGS



	A	C	B	E	F		G	H
Nominal Pipe Size	Center to End, Elbows, Tees and Crosses	Center to End, 45 Deg Elbows	Length of Thread Min.	Width of Band, Min.	Inside Diameter of Fitting Min.	Inside Diameter of Fitting Max.	Metal Thickness, Min.	Outside Diameter of Band, Min.
¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
5	4.50	3.05	1.18	1.18	5.563	5.663	0.380	7.05
6	5.13	3.46	1.28	1.28	6.625	6.725	0.430	8.28
8	6.56	4.28	1.47	1.47	8.625	8.725	0.550	10.63
10	8.08	5.16	1.68	1.68	10.750	10.850	0.690	13.12
12	9.50	5.97	1.88	1.88	12.750	12.850	0.800	15.47
14 O.D.	10.40	....	2.00	2.00	14.000	14.100	0.880	16.94
16 O.D.	11.82	....	2.20	2.20	16.000	16.100	1.000	19.30

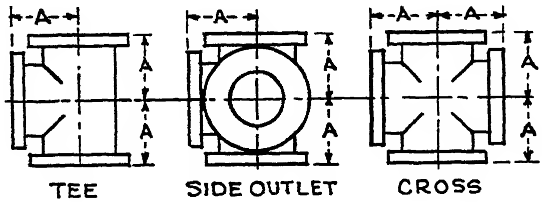
All dimensions given in inches.

CORROSION

Experiments in research laboratories have demonstrated that the amount of corrosion found in piping and closed water systems is almost directly proportional to the amount of oxygen in solution, and varies with the temperature and rate of flow. The composition of the water also has a bearing on the rate of attack. All reliable data on this subject indicate that the composition of the iron or steel as regards carbon, phosphorous,

manganese, sulphur, silicon, and copper makes very little difference in the amount or character of corrosion when under water, although under atmospheric exposure the influence of composition is more marked. Corrosion is stimulated by scale adhering to the iron surface, by electric

TABLE 6. AMERICAN STANDARD DIMENSIONS OF TEES AND CROSSES (STRAIGHT SIZES) FOR 125 LB CAST-IRON FLANGED FITTINGS



Nominal Pipe Size [a-b]	*A* Center to Face Tees and Crosses [b-c]	*AA* Face to Face Tees and Crosses [b-c]	Diameter of Flange	Thickness of Flange, Min.	Metal Thickness of Body, Min.
1	3½	7	4¼	7/16	7/16
1¼	3¾	7½	4⅝	½	7/16
1½	4	8	5	9/16	7/16
2	4½	9	6	⅝	7/16
2½	5	10	7	11/16	7/16
3	5½	11	7½	¾	7/16
3½	6	12	8½	13/16	7/16
4	6½	13	9	15/16	½
5	7½	15	10	15/16	½
6	8	16	11	1	9/16
8	9	18	13½	1⅛	⅝
10	11	22	16	1 3/16	¾
12	12	24	19	1¼	13/16
14 O.D.	14	28	21	1⅜	⅞
16 O.D.	15	30	23½	1 7/16	1
18 O.D.	16½	33	25	1 9/16	1 1/16
20 O.D.	18	36	27½	1 11/16	1⅛
24 O.D.	22	44	32	1⅞	1¼
30 O.D.	25	50	38¾	2⅛	1 7/16
36 O.D.	28	56	46	2⅜	1⅝
42 O.D.	31	62	53	2⅝	1 13/16
48 O.D.	34	68	59½	2¾	2

All dimensions given in inches.

[a]Size of all fittings listed indicates nominal inside diameter of port.

[b]Tees, side outlet tees, and crosses, 16 in. and smaller, reducing on the outlet, have the same dimensions center to face, and face to face as straight size fittings, corresponding to the size of the larger opening. Sizes 18 in. and larger, reducing on the outlet, are made in two lengths, depending on the size of the outlet.

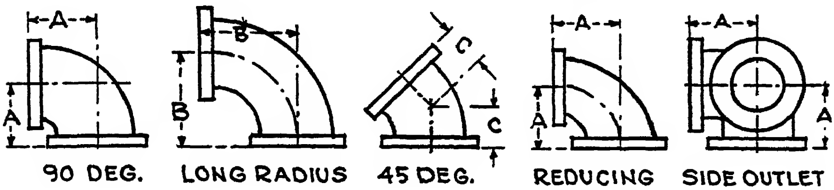
[c]Tees and crosses, reducing on run only, carry same dimensions center to face and face to face as a straight size fitting of the larger opening.

currents from external sources, by acids carried in solution or as gases, by solids that break down in water, by strains due to inadequate annealing and by contact between dissimilar materials.

The best way to deal with corrosive water supply is to treat it at its source. In practice, oxygen removal has been accomplished in two ways,

(1) by deaërating the water mechanically and (2) by fixing the free oxygen by chemical combination. Corrosive action may be retarded by the use of a metal which is electro-positive to that to be protected, such as zinc

TABLE 7. AMERICAN STANDARD DIMENSIONS OF ELBOWS FOR 125 LB CAST-IRON FLANGED FITTINGS



Nominal Pipe Size [a]	A	B	C	Diameter of Flange	Thickness of Flange, Min.	Metal Thickness of Body, Min.
	Center to Face Elbow b-c-d	Center to Face Long Radius Elbow b-c-d	Center to Face 45 Deg Elbow c			
1	3½	5	1¾	4¼	7/16	7/16
1¼	3¾	5½	2	4⅝	½	7/16
1½	4	6	2¼	5	9/16	7/16
2	4½	6½	2½	6	⅝	7/16
2½	5	7	3	7	11/16	7/16
3	5½	7¾	3	7½	¾	7/16
3½	6	8½	3½	8½	13/16	7/16
4	6½	9	4	9	15/16	½
5	7½	10¼	4½	10	15/16	½
6	8	11½	5	11	1	9/16
8	9	14	5½	13½	1⅛	⅝
10	11	16½	6½	16	1 3/16	¾
12	12	19	7½	19	1¼	13/16
14 O.D.	14	21½	7½	21	1⅜	⅞
16 O.D.	15	24	8	23½	1 7/16	1
18 O.D.	16½	26½	8½	25	1 9/16	1 1/16
20 O.D.	18	29	9½	27½	1 11/16	1⅛
24 O.D.	22	34	11	32	1⅞	1¼
30 O.D.	25	41½	15	38¾	2⅛	1 7/16
36 O.D.	28	49	18	46	2⅜	1⅝
42 O.D.	31	56½	21	53	2⅝	1 13/16
48 O.D.	34	64	24	59½	2¾	2

All dimensions given in inches.

[a] Size of all fittings listed indicates nominal inside diameter of port.

[b] Reducing elbows and side outlet elbows carry same dimensions center to face as straight size elbows, corresponding to the size of the larger opening.

[c] Special degree elbows, ranging from 1 to 45 deg, inclusive, have the same center to face dimensions as given for 45 deg elbows and those over 45 deg and up to 90 deg, inclusive, shall have the same center to face dimensions as given for 90 deg elbows. The angle designation of an elbow is its deflection from straight line flow and is the angle between the flange faces.

[d] Side outlet elbows shall have all openings on intersection center-lines.

to protect iron; by the effective neutralization of acid or alkaline contents; by the elimination of ununiform materials; and through the exercise of care in counteracting the effects of stray electric currents.

The introduction into piping systems of various chemical compounds, usually for forming a protective coating of thin deposit, has apparently

had some success. Non-ferrous piping may deteriorate rapidly unless selected with due regard to the service conditions to which it will be subjected. The composition and micro-structure are items of special importance. Electrolysis set up by joining copper alloys to iron must be guarded against.

Piping exposed to the elements or buried in the ground is quite successfully protected by coatings of the asphaltic type which are usually applied hot and are often reinforced with fabric wrappings. Galvanizing by the hot-dip process and painting with specially prepared mixtures also afford satisfactory protective safeguards.

The problem of corrosion has been summarized[2] as follows:

1. Corrosion in steam heating systems is confined mainly to the returns and varies widely in different localities and sometimes in the same locality, due to variations in the water and in operating conditions.

2. There does not seem to be any necessity to use a more costly material for piping than wrought iron or steel, between which long experience indicates no material difference in durability when used for this purpose.

3. The amount of dissolved oxygen and carbon dioxide present determines the extent of corrosion. The carbon dioxide should be low, especially when any considerable amount of oxygen is present.

4. The carbon dioxide in the condensate comes mostly from the dissociation of bicarbonates or carbonates in the boiler. For this reason sodium carbonate should not be used for internal treatment of water in heating boilers.

5. In high pressure plants the water should be pretreated to remove free and half-bound CO_2 and scale-forming matter without leaving more than a slight excess of sodium carbonate. The water should be thoroughly deaerated before entering the boiler.

6. In low pressure steam boilers (less than 5 lb pressure) where corrosion occurs in the returns, an excess of caustic soda with an equal amount of sodium sulphate should be maintained in the boiling water. The hydroxyl alkalinity should be at least twice the carbonate alkalinity. This may be controlled by testing samples of boiler water and the condensate.

7. All returns should be sealed, and every precaution taken to prevent unnecessary leakage of air into the system. Oiling the steam will afford substantial protection to steam piping.

A series of investigations, on the factors influencing corrosion in steam systems, including a study of the raw water, the production, the distribution, and the utilization of steam, demonstrated the extent and constitution of deposits in the heating systems of a large office building and a large hotel in New York City[3]. These deposits were found to originate from the action of corrosion and were found to contain little material from the boilers. The cause of the excessive corrosion was found to be the in-leakage of air into the vacuum return system. The amounts of oxygen and carbon dioxide associated with the steam were shown to be of insignificant importance as corrosive agents by the application of the Law of Henry and Dalton. Causes of corrosion trouble in heating systems were reported in the operation and, to a small extent, in the design of the system and not in the quality of steam used if that quality equals that encountered in these studies.

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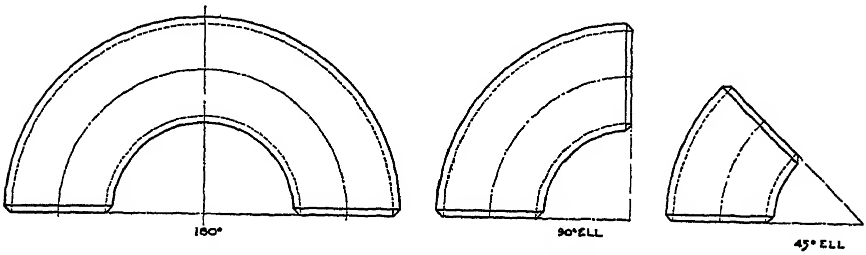
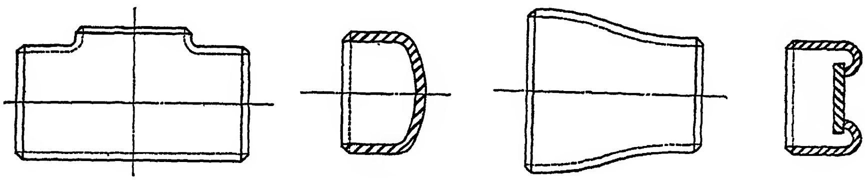
[2]Corrosion in Steam Heating Systems, by F. N. Speller (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928).

[3]Some Fundamental Considerations of Corrosion in Steam and Condensate Lines, by R. E. Hall and A. R. Mumford (A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932).

A study of water supply as related to *air conditioning equipment*[4] indicated that the characteristics of waters used vary widely. Certain waters are corrosive while others are scale forming[5]. Waters often bring about high rates of depreciation of apparatus and lower operating efficiencies. Chemical analysis of waters followed by chemical conditioning may obviate many of the difficulties encountered.



a. TYPICAL SHORT RADIUS ELBOWS



b. TEE *c.* FORGED CAP *d.* CONCENTRIC REDUCER *e.* END CLOSURE

f. WELDING NECK FLANGE *g.* LAP JOINT FLANGED WELDING NECK

FIG. 2. WELDING FITTINGS

PIPE WELDING

Fusion welding in the heating and ventilating industry is an improved method of fabrication and installation widely used and accepted as competitive to the screwed and flanged joint in piping, and the rivet and bolt assembly in sheet and plate construction.

The process of fusion welding is easily understood. Two pieces of metal are brought substantially together and melted, with the addition of filler

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[4]Corrosion as related to Air Conditioning Equipment, by R. M. Palmer.

[5]For additional information on corrosion, see 1933 report of *National District Heating Association* on this subject.

rod, by the welding flame. The molten metal will flow together with the aid of mechanical manipulation by the workman so that when cooled there is a single continuous unit. Welding application is made by either the oxy-acetylene or electric arc processes, and the two processes are given equal rating when used under proper control standards and intelligent supervision.

The welding process is applied with equal efficiency and economy in high and low pressure service, and thru the range of all pipe sizes. A correct understanding of the diversified application of welding will in most cases reflect lower initial costs and complete elimination of maintenance. Reduction in weight, adaptability to space allowance, saving in supplemental materials such as pipe covering, hangers, and supports, and finished appearance are other distinct advantages in favor of welding which have contributed to its wide and economical use.

Welding application requires the same basic knowledge of design as do the other types of assembly, but in addition, requires a generous knowledge of the sciences involved, particularly as to welding qualities of metal, their reaction to extremely high temperatures, and the ability to determine and use only the best quality welding rods. This requirement applies equally to employer and employee with the employer accepting all of the responsibility. Thus the employer should select his welding mechanics with good judgment, provide them with first-class equipment and tools, arrange for their training and use of acceptable workmanship standards, and at regular intervals subject their work to prescribed tests. Industry will not accept the employment of mechanics of undetermined ability nor on the basis of past experience. Neither does industry accept the statement that a weld is only as good as the workman who makes it. The control Codes now in process of adoption will be the law governing the use of the welding process. These Codes prohibit individual practices contrary to their specified procedure and rules of control, and this is predicated upon the sound requirement that the employer must assume full responsibility for the deposited weld.

It is advisable that this management responsibility be included in all welding specifications and that authoritative standards of workmanship also be specified. The standards of workmanship for this industry are as set forth in the *Standard Manual on Pipe Welding* of the *Heating, Piping and Air Conditioning Contractors National Association.*

A complete line of manufactured steel welding fittings is now available with plain ends machine beveled for welding and with radii similar to short and long radius flanged fittings. Some typical types of these fittings are shown in Fig. 2. They are made in pipe sizes ¾ to 24 in., standard and extra heavy, in steel, wrought iron, brass, copper, and special alloys.

Chapter 35

HEAT LOSSES FROM BARE AND INSULATED PIPES

Heat Losses from Bare Pipes, Steam and Hot Water Lines, Low Temperature Pipe Insulation, Pipe Sweating, Heat Losses from Pipe Surfaces, Thickness of Pipe Insulation, Underground Insulation

WHEN steam or hot water are conveyed from one part of a building to another, it usually is desirable that the loss of heat from the pipes through which these heating media pass be reduced to an economic minimum by means of the proper types and thicknesses of insulation. Pipe insulation is also used to reduce the absorption of heat by cold pipes as well as to prevent condensation on the outer surfaces.

HEAT LOSSES FROM BARE PIPES

Heat losses from horizontal bare iron pipes, based on data obtained from tests conducted at the Mellon Institute, are given in Table 1. These losses are expressed in Btu per hour per linear foot of pipe per degree Fahrenheit difference in temperature between the steam or hot water in the pipe and the air surrounding the pipe. The monetary value of the loss of heat given in Table 1 may be obtained by means of Fig. 1 for various heating system efficiencies, temperature differences and calorific values and costs of coal. To solve a problem, select the proper heat loss coefficient from Table 1 and locate this value on the upper left hand margin of the chart. Then draw lines in the order indicated by the dotted lines, the dollar value of the heat loss per 100 linear feet of pipe per 1000 hours being given on the upper right hand scale. In using this chart, the cost of coal should also include the labor for handling it, boiler room expense, etc. For additional information on this subject refer to paper entitled Heat Emission from Iron and Copper Pipe[1], by F. C. Houghten and Carl Gutberlet.

In order to determine heat losses per linear foot of pipe from known losses per square foot, it is necessary to know the area in square feet per linear foot of pipe. Table 2 gives these areas for various standard pipe sizes while Table 3 gives the area in square feet for flanges and fittings for various standard pipe sizes.

Very often, even where pipes are thoroughly insulated, flanges and fittings are left bare due to the belief that the losses from these parts are not large. However, the fact that a pair of 9-in. standard flanges having an area of 3.00 sq ft would lose, at 100 lb steam pressure, an amount of

[1]A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932.

TABLE 1. HEAT LOSSES FROM HORIZONTAL BARE IRON PIPES

Expressed in Btu per linear foot per degree Fahrenheit difference in temperature between the pipe and surrounding still air at 70 F

Nominal Pipe Size (Inches)	Hot Water				Steam		
	120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
	Temperature Difference						
	50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
½	0.543	0.573	0.605	0.638	0.656	0.742	0.796
¾	0.660	0.690	0.729	0.762	0.781	0.886	0.955
1	0.791	0.829	0.878	0.920	0.953	1.084	1.166
1¼	0.979	1.02	1.087	1.15	1.184	1.345	1.450
1½	1.09	1.15	1.220	1.29	1.335	1.520	1.640
2	1.34	1.40	1.491	1.58	1.637	1.866	2.015
2½	1.58	1.67	1.778	1.87	1.937	2.215	2.388
3	1.88	1.99	2.100	2.22	2.301	2.641	2.853
3½	2.13	2.24	2.380	2.51	2.585	2.972	3.215
4	2.36	2.50	2.650	2.78	2.873	3.312	3.582
4½	2.60	2.75	2.920	3.08	3.170	3.655	3.956
5	2.87	3.02	3.200	3.38	3.493	4.030	4.368
6	3.39	3.56	3.775	4.01	4.115	4.755	5.153
8	4.32	4.55	5.050	5.14	5.270	6.120	6.635
9	4.80	5.05	5.350	5.71	5.885	6.824	7.400
12	6.25	6.62	6.995	7.46	7.670	8.900	9.670

TABLE 2. RADIATING SURFACE PER LINEAR FOOT OF PIPE

Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)	Nominal Pipe Size (Inches)	Surface Area (Sq Ft)
½	0.22	2	0.622	5	1.456
¾	0.275	2½	0.753	6	1.734
1	0.344	3	0.917	8	2.257
1¼	0.435	3½	1.047	9	2.519
1½	0.498	4	1.178	12	3.338

TABLE 3. AREAS OF FLANGED FITTINGS, SQUARE FEET[a]

Nominal Pipe Size (Inches)	Flanged Coupling		90 Deg Ell		Long Radius Ell		Tee		Cross	
	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy	Standard	Extra Heavy
1	0.320	0.438	0.795	1.015	0.892	1.083	1.235	1.575	1.622	2.07
1¼	0.383	0.510	0.957	1.098	1.084	1.340	1.481	1.925	1.943	2.53
1½	0.477	0.727	1.174	1.332	1.337	1.874	1.815	2.68	2.38	3.54
2	0.672	0.848	1.65	2.01	1.84	2.16	2.54	3.09	3.32	4.06
2½	0.841	1.107	2.09	2.57	2.32	2.76	3.21	4.05	4.19	5.17
3	0.945	1.484	2.38	3.49	2.68	3.74	3.66	5.33	4.77	6.95
3½	1.122	1.644	2.98	3.96	3.28	4.28	4.48	6.04	5.83	7.89
4	1.344	1.914	3.53	4.64	3.96	4.99	5.41	7.07	7.03	9.24
4½	1.474	2.04	3.95	5.02	4.43	5.46	6.07	7.72	7.87	10.07
5	1.622	2.18	4.44	5.47	5.00	6.02	6.81	8.52	8.82	10.97
6	1.82	2.78	5.13	6.99	5.99	7.76	7.84	10.64	10.08	13.75
8	2.41	3.77	6.98	9.76	8.56	11.09	10.55	14.74	13.44	18.97
9	3.00	4.44	8.71	11.44	10.57	13.17	13.18	17.23	16.78	22.10
12	4.41	6.71	13.08	17.73	16.35	18.76	19.67	26.65	24.87	34.11

[a]Including areas of accompanying flanges bolted to the fitting.

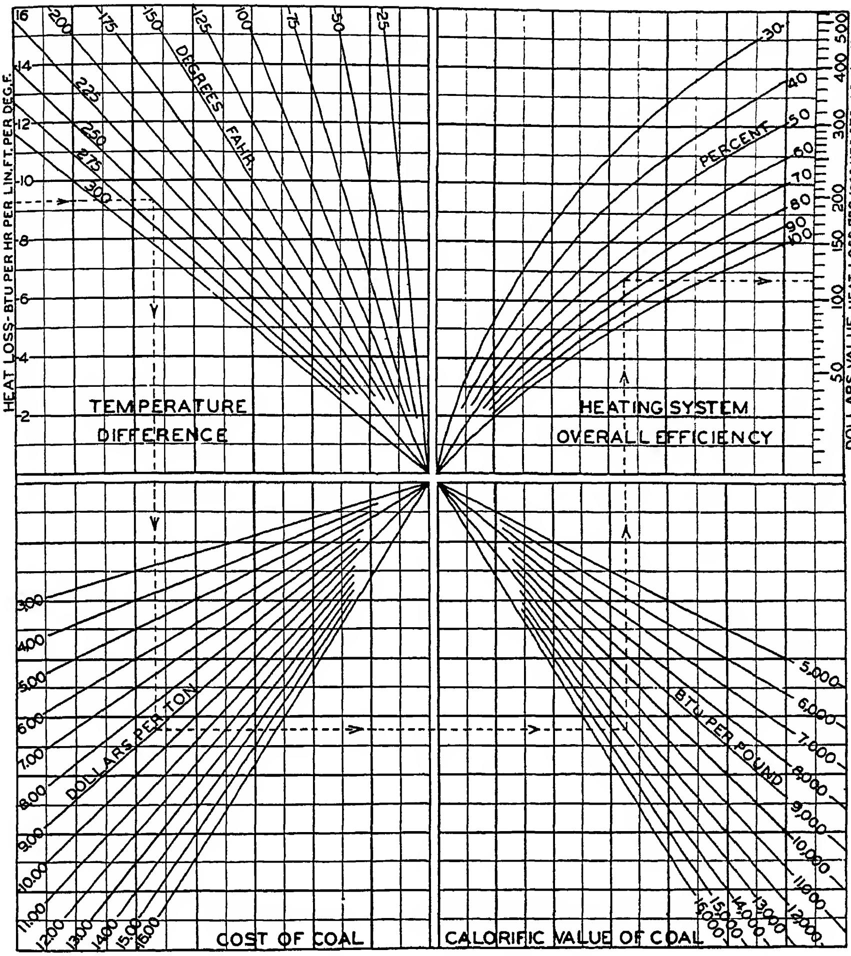


FIG. 1. CHART FOR ESTIMATING DOLLAR VALUE OF HEAT LOSS FROM BARE IRON PIPES. (SEE TABLE 1)[a]

[a]This chart is based on 100 linear feet per 1000 hours. For fractions or multiples of these factors, multiply by proper percentage.

heat equivalent to more than a ton of coal per year shows the necessity for insulating such surfaces. Table 3 shows the areas of both standard and extra heavy flanged fittings including the accompanying flanges bolted to the fittings.

STEAM AND HOT WATER LINES

The conductivities of various materials used for insulating steam and hot water pipes are given in Table 4. In this table the conductivities are given as functions of the mean temperatures or the mean of the inner and outer surface temperatures of the insulations. This method of stating

TABLE 4. CONDUCTIVITIES (k) OF VARIOUS TYPES OF INSULATING MATERIALS FOR MEDIUM AND HIGH TEMPERATURE PIPES[a]

	Mean Temperature				
	100 F	200 F	300 F	400 F	500 F
85 per cent Magnesia Type	0.425	0.465	0.505	0.550	0.590
Corrugated Asbestos Type (4 Plies per 1 in. thick)	0.530	0.650	0.770	0.890	
Corrugated Asbestos Type (8 Plies per 1 in. thick)	0.480	0.555	0.630	0.705	
Laminated Asbestos Type (30–40 Laminations per 1 in. thick)	0.360	0.415	0.470	0.525	0.585
Laminated Asbestos Type (20 Laminations per 1 in. thick)	0.545	0.605	0.665	0.725	0.785
Rock Wool Type	0.350	0.410	0.470	0.530	0.590
High Temperature Type (Diatomaceous Earth and Asbestos)	0.515	0.545	0.575	0.605	0.635
Brown Asbestos Type (Felted Fibre)	0.600	0.640	0.675	0.715	0.750

[a]*Mechanical Engineer's Handbook*, Marks, 3rd Ed., 1930.

conductivities makes it possible readily to calculate the heat loss through single or compound sections. It should be emphasized that the conductivities given in Table 4 for the various insulations are the average of values obtained from a number of tests made on each type of material, also that all variables due to differences in thickness, pipe sizes, and air conditions are eliminated. Individual manufacturer's materials will, of course, vary in conductivity to some extent from these values.

The heat losses through six of the types of insulation given in Table 4 for 1, 1½ and 2-in.-thick materials, and for temperatures commonly encountered in engineering practice can be obtained from Tables 5 to 10, inclusive. The loss through other thicknesses of the materials, and for other hot water or steam temperature conditions may be obtained by interpolation. The heat loss coefficients given in Tables 5 to 10 are based on the conductivities in Table 4 and were computed from data given in Chapter 22, THE GUIDE 1931, as in the following problem:

Example 1. Determine the total heat loss for a period of 30 days through a 1-in. (4 ply) thickness of corrugated asbestos insulation on 100 ft of 3-in. pipe carrying steam at 5 lb pressure, and with surrounding air at 60 F. Also determine percentage of heat saving over bare pipe loss.

Solution. Difference in temperature between pipe and surrounding air = 227.1 − 60 or 167.1 deg. From Table 6 the coefficients of transmission for 1-in. corrugated asbestos (4 ply) are 0.608 and 0.652 Btu per hour per square foot per degree temperature difference at temperature differences of 157.1 deg and 227.7 deg, respectively. The difference is 0.044 Btu per hour per square foot per degree temperature difference for a temperature difference of 70.5 deg. For a temperature difference of 167.1 deg (10 deg higher than 157.1 deg) the coefficient of transmission is then $0.608 + \left(\frac{10}{70.5} \times 0.044\right)$, or 0.614 Btu per hour per square foot per degree temperature difference. From Table 2 the area per linear foot of 3-in. pipe is 0.917 sq ft. The total heat loss is therefore 0.917 × 0.614 × 167.1 × 100 (linear feet) × 720 (hours) = 6,774,000 Btu.

From Table 1, and in the same manner it is determined that the loss through an uninsulated 3-in. pipe for the same conditions will be 25,917,000 Btu. The saving due to insulation is 19,143,000 Btu or 74 per cent of the bare pipe loss.

Table 5. Coefficients of Transmission (U) for Pipes Insulated with 85 Per Cent Magnesia Type Insulation

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.744	0.754	0.764	0.774	0.779	0.802	0.814
	¾	0.672	0.681	0.689	0.697	0.701	0.721	0.731
	1	0.613	0.621	0.629	0.637	0.641	0.659	0.670
	1¼	0.562	0.570	0.577	0.585	0.589	0.606	0.617
	1½	0.532	0.539	0.546	0.553	0.557	0.573	0.582
	2	0.500	0.506	0.512	0.519	0.523	0.538	0.547
	2½	0.475	0.481	0.487	0.493	0.497	0.512	0.520
	3	0.455	0.461	0.467	0.474	0.477	0.492	0.500
	3½	0.441	0.447	0.452	0.458	0.462	0.475	0.483
	4	0.429	0.435	0.441	0.446	0.449	0.463	0.471
	4½	0.420	0.425	0.431	0.437	0.440	0.453	0.460
	5	0.411	0.416	0.422	0.427	0.430	0.443	0.450
	6	0.402	0.408	0.413	0.419	0.422	0.435	0.442
	8	0.387	0.392	0.397	0.403	0.405	0.418	0.425
	9	0.380	0.385	0.390	0.395	0.398	0.410	0.417
	12	0.369	0.374	0.378	0.383	0.386	0.398	0.405
1½	½	0.617	0.625	0.633	0.642	0.646	0.665	0.676
	¾	0.550	0.558	0.566	0.573	0.577	0.596	0.606
	1	0.496	0.503	0.511	0.518	0.522	0.540	0.549
	1¼	0.453	0.459	0.465	0.472	0.475	0.490	0.498
	1½	0.424	0.430	0.436	0.442	0.445	0.459	0.467
	2	0.394	0.400	0.405	0.410	0.413	0.427	0.434
	2½	0.371	0.376	0.382	0.386	0.389	0.401	0.408
	3	0.352	0.357	0.362	0.367	0.370	0.380	0.387
	3½	0.339	0.343	0.347	0.351	0.354	0.364	0.370
	4	0.328	0.333	0.337	0.341	0.343	0.353	0.359
	4½	0.320	0.324	0.328	0.332	0.334	0.343	0.350
	5	0.312	0.316	0.320	0.324	0.326	0.336	0.342
	6	0.303	0.307	0.311	0.315	0.318	0.328	0.333
	8	0.287	0.291	0.295	0.299	0.301	0.311	0.316
	9	0.280	0.284	0.288	0.292	0.294	0.303	0.308
	12	0.272	0.275	0.279	0.283	0.285	0.294	0.299
2	½	0.543	0.551	0.558	0.565	0.569	0.587	0.597
	¾	0.484	0.490	0.497	0.503	0.507	0.523	0.532
	1	0.433	0.439	0.445	0.451	0.454	0.467	0.476
	1¼	0.393	0.398	0.403	0.409	0.412	0.424	0.432
	1½	0.365	0.370	0.376	0.381	0.384	0.397	0.402
	2	0.338	0.343	0.347	0.351	0.354	0.364	0.370
	2½	0.316	0.320	0.324	0.328	0.331	0.341	0.347
	3	0.297	0.301	0.305	0.309	0.312	0.321	0.326
	3½	0.284	0.288	0.292	0.295	0.297	0.306	0.311
	4	0.275	0.278	0.282	0.285	0.287	0.296	0.301
	4½	0.266	0.270	0.273	0.276	0.278	0.286	0.290
	5	0.258	0.262	0.265	0.268	0.270	0.278	0.283
	6	0.250	0.254	0.257	0.260	0.262	0.270	0.274
	8	0.236	0.239	0.242	0.245	0.247	0.255	0.258
	9	0.228	0.231	0.234	0.237	0.239	0.246	0.250
	12	0.219	0.222	0.225	0.228	0.230	0.237	0.240

Table 6. Coefficients of Transmission (U) for Pipes Insulated with Corrugated Asbestos Type Insulation (4 Plies per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.890	0.919	0.949	0.978	0.995	1.065	1.106
	¾	0.803	0.829	0.857	0.883	0.898	0.961	0.997
	1	0.731	0.756	0.780	0.804	0.818	0.876	0.909
	1¼	0.671	0.693	0.716	0.738	0.751	0.804	0.834
	1½	0.635	0.656	0.677	0.698	0.710	0.760	0.788
	2	0.595	0.615	0.635	0.656	0.667	0.715	0.742
	2½	0.567	0.586	0.605	0.624	0.635	0.680	0.705
	3	0.544	0.562	0.580	0.598	0.608	0.652	0.677
1	3½	0.527	0.544	0.561	0.578	0.588	0.631	0.654
	4	0.513	0.530	0.548	0.565	0.575	0.616	0.639
	4½	0.502	0.518	0.535	0.551	0.561	0.601	0.624
	5	0.490	0.507	0.523	0.539	0.549	0.588	0.611
	6	0.480	0.496	0.512	0.528	0.538	0.577	0.599
	8	0.462	0.477	0.493	0.508	0.517	0.554	0.575
	9	0.453	0.468	0.483	0.498	0.507	0.544	0.564
	12	0.441	0.456	0.470	0.485	0.493	0.529	0.550
	½	0.737	0.762	0.787	0.812	0.826	0.884	0.918
	¾	0.657	0.679	0.702	0.725	0.737	0.790	0.820
	1	0.594	0.614	0.634	0.654	0.666	0.713	0.740
	1¼	0.542	0.559	0.577	0.596	0.606	0.649	0.673
	1½	0.507	0.524	0.541	0.558	0.568	0.609	0.632
	2	0.471	0.487	0.503	0.519	0.528	0.565	0.587
	2½	0.443	0.458	0.473	0.488	0.497	0.533	0.553
	3	0.421	0.435	0.449	0.463	0.472	0.506	0.525
1½	3½	0.403	0.417	0.430	0.443	0.451	0.483	0.502
	4	0.393	0.405	0.418	0.432	0.439	0.471	0.489
	4½	0.383	0.394	0.407	0.420	0.428	0.460	0.476
	5	0.372	0.384	0.397	0.409	0.417	0.447	0.463
	6	0.362	0.374	0.387	0.399	0.406	0.436	0.452
	8	0.343	0.354	0.366	0.378	0.385	0.413	0.429
	9	0.334	0.345	0.357	0.368	0.375	0.403	0.419
	12	0.323	0.334	0.346	0.357	0.364	0.391	0.407
	½	0.648	0.670	0.692	0.713	0.726	0.779	0.810
	¾	0.578	0.598	0.617	0.637	0.648	0.694	0.720
	1	0.518	0.535	0.552	0.570	0.580	0.622	0.645
	1¼	0.469	0.485	0.501	0.517	0.527	0.566	0.587
	1½	0.438	0.452	0.467	0.481	0.490	0.526	0.545
	2	0.404	0.417	0.430	0.444	0.452	0.483	0.502
	2½	0.379	0.391	0.403	0.415	0.422	0.451	0.466
	3	0.356	0.367	0.378	0.390	0.397	0.425	0.440
2	3½	0.339	0.350	0.361	0.373	0.380	0.406	0.421
	4	0.328	0.339	0.350	0.360	0.367	0.392	0.406
	4½	0.318	0.328	0.339	0.350	0.357	0.381	0.395
	5	0.308	0.318	0.329	0.340	0.346	0.370	0.384
	6	0.299	0.309	0.319	0.329	0.335	0.358	0.371
	8	0.282	0.291	0.301	0.310	0.315	0.336	0.349
	9	0.273	0.282	0.291	0.300	0.305	0.325	0.338
	12	0.263	0.272	0.280	0.289	0.294	0.314	0.325

Table 7. Coefficients of Transmission (U) for Pipes Insulated with Corrugated Asbestos Type Insulation (8 Plies per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.801	0.820	0.838	0.857	0.868	0.913	0.939
	¾	0.723	0.739	0.756	0.773	0.783	0.824	0.847
	1	0.658	0.673	0.688	0.704	0.713	0.751	0.772
	1¼	0.606	0.619	0.633	0.647	0.655	0.688	0.707
	1½	0.573	0.586	0.599	0.612	0.619	0.652	0.670
	2	0.538	0.550	0.562	0.575	0.581	0.612	0.629
	2½	0.511	0.523	0.534	0.546	0.553	0.582	0.599
	3	0.489	0.501	0.512	0.524	0.531	0.558	0.575
1	3½	0.474	0.485	0.496	0.507	0.514	0.542	0.557
	4	0.461	0.472	0.482	0.493	0.500	0.527	0.542
	4½	0.451	0.462	0.472	0.482	0.489	0.515	0.530
	5	0.442	0.452	0.462	0.473	0.479	0.505	0.520
	6	0.432	0.442	0.452	0.463	0.468	0.493	0.508
	8	0.416	0.426	0.436	0.446	0.451	0.475	0.489
	9	0.408	0.418	0.427	0.437	0.442	0.466	0.480
	12	0.397	0.406	0.415	0.424	0.429	0.452	0.466
	½	0.664	0.679	0.695	0.711	0.720	0.759	0.780
	¾	0.593	0.607	0.621	0.636	0.643	0.677	0.697
	1	0.535	0.547	0.560	0.573	0.580	0.611	0.629
	1¼	0.488	0.499	0.510	0.522	0.528	0.556	0.572
	1½	0.457	0.467	0.478	0.490	0.496	0.522	0.537
	2	0.425	0.434	0.444	0.455	0.460	0.485	0.499
	2½	0.399	0.408	0.418	0.428	0.434	0.457	0.471
	3	0.378	0.387	0.396	0.405	0.411	0.433	0.446
1½	3½	0.363	0.371	0.380	0.388	0.393	0.415	0.427
	4	0.353	0.361	0.369	0.378	0.383	0.403	0.415
	4½	0.343	0.351	0.360	0.368	0.373	0.393	0.404
	5	0.334	0.342	0.350	0.358	0.363	0.383	0.394
	6	0.325	0.333	0.341	0.349	0.353	0.373	0.383
	8	0.309	0.316	0.324	0.332	0.336	0.355	0.365
	9	0.301	0.309	0.316	0.324	0.328	0.346	0.356
	12	0.291	0.298	0.306	0.313	0.317	0.335	0.344
	½	0.585	0.599	0.613	0.627	0.635	0.668	0.688
	¾	0.520	0.533	0.545	0.558	0.565	0.595	0.612
	1	0.465	0.476	0.487	0.498	0.504	0.532	0.547
	1¼	0.422	0.432	0.442	0.452	0.458	0.483	0.497
	1½	0.394	0.403	0.412	0.422	0.427	0.450	0.462
	2	0.364	0.372	0.380	0.388	0.393	0.415	0.427
	2½	0.339	0.347	0.355	0.363	0.367	0.387	0.398
	3	0.319	0.327	0.334	0.342	0.346	0.365	0.375
2	3½	0.304	0.311	0.318	0.326	0.330	0.349	0.358
	4	0.295	0.302	0.308	0.315	0.319	0.336	0.345
	4½	0.285	0.292	0.299	0.306	0.310	0.327	0.336
	5	0.278	0.284	0.290	0.297	0.301	0.317	0.326
	6	0.269	0.275	0.282	0.288	0.292	0.307	0.315
	8	0.253	0.259	0.265	0.270	0.273	0.288	0.296
	9	0.244	0.250	0.256	0.262	0.265	0.280	0.287
	12	0.236	0.241	0.247	0.253	0.256	0.270	0.277

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.605	0.620	0.635	0.650	0.658	0.695	0.716
	¾	0.546	0.560	0.573	0.586	0.594	0.627	0.645
	1	0.498	0.510	0.522	0.534	0.541	0.570	0.587
	1¼	0.457	0.468	0.480	0.491	0.497	0.525	0.540
	1½	0.432	0.442	0.453	0.464	0.470	0.496	0.511
	2	0.406	0.416	0.426	0.437	0.442	0.467	0.481
	2½	0.385	0.395	0.405	0.415	0.420	0.443	0.457
	3	0.370	0.379	0.389	0.398	0.403	0.425	0.438
	3½	0.359	0.367	0.376	0.385	0.390	0.413	0.426
	4	0.349	0.358	0.366	0.375	0.380	0.402	0.414
	4½	0.341	0.350	0.359	0.367	0.372	0.393	0.405
	5	0.334	0.342	0.351	0.359	0.364	0.384	0.395
	6	0.327	0.335	0.343	0.351	0.356	0.376	0.387
	8	0.314	0.322	0.330	0.338	0.343	0.362	0.373
	9	0.308	0.316	0.325	0.333	0.336	0.355	0.366
	12	0.301	0.308	0.316	0.324	0.328	0.346	0.356
1½	½	0.502	0.514	0.526	0.539	0.546	0.577	0.595
	¾	0.450	0.461	0.473	0.484	0.490	0.517	0.532
	1	0.405	0.415	0.426	0.436	0.442	0.466	0.480
	1¼	0.369	0.378	0.387	0.396	0.401	0.423	0.435
	1½	0.343	0.352	0.361	0.370	0.375	0.397	0.409
	2	0.321	0.329	0.337	0.345	0.350	0.369	0.380
	2½	0.301	0.309	0.317	0.324	0.330	0.348	0.358
	3	0.286	0.293	0.301	0.308	0.313	0.330	0.340
	3½	0.274	0.281	0.288	0.295	0.300	0.316	0.326
	4	0.267	0.273	0.280	0.287	0.291	0.307	0.317
	4½	0.259	0.266	0.272	0.279	0.283	0.299	0.308
	5	0.253	0.260	0.266	0.272	0.276	0.291	0.300
	6	0.247	0.253	0.260	0.266	0.269	0.284	0.293
	8	0.234	0.240	0.246	0.252	0.255	0.270	0.279
	9	0.227	0.233	0.239	0.245	0.249	0.263	0.271
	12	0.221	0.227	0.232	0.238	0.241	0.255	0.263
2	½	0.442	0.453	0.464	0.475	0.481	0.508	0.523
	¾	0.392	0.402	0.412	0.422	0.428	0.452	0.465
	1	0.352	0.360	0.369	0.378	0.383	0.405	0.417
	1¼	0.319	0.327	0.335	0.343	0.348	0.367	0.379
	1½	0.297	0.304	0.311	0.319	0.323	0.341	0.352
	2	0.274	0.280	0.287	0.294	0.298	0.314	0.324
	2½	0.256	0.262	0.269	0.275	0.279	0.293	0.302
	3	0.243	0.249	0.254	0.260	0.264	0.277	0.285
	3½	0.231	0.236	0.242	0.248	0.251	0.265	0.273
	4	0.223	0.228	0.234	0.240	0.243	0.257	0.265
	4½	0.216	0.222	0.227	0.233	0.236	0.249	0.256
	5	0.210	0.215	0.220	0.225	0.228	0.241	0.248
	6	0.203	0.208	0.213	0.218	0.221	0.233	0.240
	8	0.191	0.196	0.201	0.206	0.209	0.220	0.227
	9	0.185	0.190	0.195	0.200	0.203	0.214	0.220
	12	0.178	0.183	0.187	0.192	0.195	0.205	0.210

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
1	½	0.910	0.925	0.940	0.956	0.964	1.001	1.022
	¾	0.823	0.836	0.850	0.863	0.871	0.902	0.921
	1	0.748	0.760	0.773	0.785	0.792	0.823	0.840
	1¼	0.686	0.698	0.710	0.721	0.728	0.756	0.771
	1½	0.649	0.659	0.671	0.682	0.688	0.716	0.731
	2	0.610	0.620	0.630	0.640	0.647	0.671	0.685
	2½	0.581	0.590	0.600	0.609	0.615	0.638	0.651
	3	0.558	0.567	0.576	0.585	0.591	0.613	0.626
	3½	0.539	0.548	0.557	0.566	0.571	0.592	0.604
	4	0.524	0.532	0.541	0.551	0.556	0.577	0.589
	4½	0.514	0.522	0.530	0.539	0.544	0.564	0.575
	5	0.503	0.511	0.519	0.528	0.533	0.553	0.565
	6	0.492	0.500	0.509	0.517	0.522	0.542	0.553
	8	0.473	0.480	0.488	0.497	0.502	0.521	0.532
	9	0.463	0.471	0.478	0.486	0.491	0.510	0.520
	12	0.452	0.459	0.467	0.475	0.478	0.497	0.507
1½	½	0.755	0.767	0.780	0.793	0.800	0.831	0.848
	¾	0.674	0.685	0.697	0.708	0.715	0.743	0.759
	1	0.607	0.618	0.628	0.639	0.645	0.670	0.684
	1¼	0.553	0.562	0.572	0.581	0.587	0.610	0.622
	1½	0.517	0.527	0.536	0.545	0.550	0.572	0.584
	2	0.481	0.490	0.499	0.508	0.513	0.535	0.547
	2½	0.453	0.460	0.469	0.477	0.481	0.500	0.511
	3	0.429	0.436	0.444	0.452	0.456	0.475	0.485
	3½	0.412	0.419	0.427	0.434	0.438	0.456	0.465
	4	0.400	0.407	0.415	0.422	0.426	0.443	0.453
	4½	0.390	0.396	0.402	0.409	0.413	0.429	0.437
	5	0.380	0.386	0.393	0.400	0.403	0.418	0.427
	6	0.369	0.375	0.382	0.389	0.392	0.408	0.417
	8	0.351	0.358	0.364	0.370	0.374	0.388	0.397
	9	0.342	0.349	0.355	0.361	0.364	0.378	0.387
	12	0.332	0.338	0.344	0.350	0.353	0.367	0.375
2	½	0.664	0.675	0.687	0.698	0.704	0.732	0.747
	¾	0.591	0.601	0.611	0.621	0.627	0.652	0.665
	1	0.529	0.538	0.547	0.557	0.562	0.584	0.597
	1¼	0.480	0.488	0.497	0.505	0.510	0.529	0.540
	1½	0.445	0.453	0.462	0.470	0.475	0.494	0.504
	2	0.412	0.420	0.427	0.434	0.438	0.455	0.464
	2½	0.385	0.392	0.398	0.405	0.409	0.425	0.434
	3	0.364	0.370	0.376	0.382	0.385	0.400	0.408
	3½	0.346	0.352	0.358	0.365	0.368	0.382	0.390
	4	0.336	0.342	0.348	0.354	0.357	0.371	0.378
	4½	0.325	0.332	0.338	0.343	0.346	0.360	0.367
	5	0.316	0.322	0.327	0.333	0.336	0.349	0.356
	6	0.306	0.312	0.317	0.323	0.326	0.338	0.345
	8	0.288	0.293	0.298	0.303	0.306	0.317	0.324
	9	0.280	0.284	0.289	0.294	0.297	0.307	0.313
	12	0.269	0.274	0.278	0.283	0.286	0.296	0.302

Table 8. Coefficients of Transmission (U) for Pipes Insulated with Laminated Asbestos Type Insulation (30 to 40 Laminations Per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Table 9. Coefficients of Transmission (U) for Pipes Insulated with Laminated Asbestos Type Insulation (Approximately 20 Laminations Per Inch Thickness)

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

TABLE 10. COEFFICIENTS OF TRANSMISSION (U) FOR PIPES INSULATED WITH ROCK WOOL TYPE INSULATION

These coefficients are expressed in Btu per hour per square foot of pipe surface per degree Fahrenheit difference in temperature between pipe and surrounding still air at 70 F

Thickness of Insulation (Inches)	Nominal Pipe Size (Inches)	Hot Water				Steam		
		120 F	150 F	180 F	210 F	227.1 F (5 Lb)	297.7 F (50 Lb)	337.9 F (100 Lb)
		Temperature Difference						
		50 F	80 F	110 F	140 F	157.1 F	227.7 F	267.9 F
	½	0.631	0.644	0.658	0.672	0.680	0.712	0.730
	¾	0.569	0.581	0.593	0.606	0.613	0.642	0.659
	1	0.518	0.529	0.541	0.552	0.559	0.585	0.600
	1¼	0.476	0.486	0.497	0.507	0.513	0.537	0.551
	1½	0.450	0.460	0.470	0.480	0.485	0.508	0.522
	2	0.422	0.431	0.441	0.450	0.456	0.478	0.490
	2½	0.402	0.411	0.420	0.428	0.434	0.455	0.466
	3	0.385	0.394	0.402	0.411	0.415	0.435	0.446
1	3½	0.373	0.381	0.389	0.398	0.402	0.421	0.432
	4	0.363	0.371	0.379	0.387	0.392	0.411	0.422
	4½	0.355	0.363	0.371	0.379	0.383	0.402	0.413
	5	0.348	0.356	0.364	0.371	0.376	0.394	0.404
	6	0.341	0.348	0.356	0.363	0.368	0.386	0.396
	8	0.327	0.335	0.342	0.349	0.353	0.372	0.381
	9	0.321	0.328	0.335	0.342	0.347	0.365	0.374
	12	0.313	0.320	0.327	0.334	0.338	0.355	0.364
	½	0.523	0.534	0.545	0.556	0.563	0.590	0.606
	¾	0.468	0.477	0.487	0.497	0.503	0.528	0.542
	1	0.421	0.430	0.440	0.449	0.455	0.477	0.490
	1¼	0.383	0.391	0.399	0.407	0.412	0.433	0.444
	1½	0.359	0.366	0.375	0.383	0.387	0.407	0.419
	2	0.333	0.340	0.348	0.356	0.360	0.378	0.389
	2½	0.314	0.320	0.327	0.335	0.339	0.355	0.365
	3	0.296	0.302	0.310	0.317	0.321	0.337	0.347
1½	3½	0.286	0.291	0.298	0.304	0.307	0.323	0.332
	4	0.278	0.284	0.290	0.296	0.300	0.315	0.323
	4½	0.270	0.276	0.282	0.287	0.291	0.305	0.313
	5	0.263	0.269	0.275	0.280	0.284	0.298	0.305
	6	0.257	0.262	0.267	0.273	0.277	0.290	0.297
	8	0.244	0.249	0.254	0.260	0.263	0.276	0.283
	9	0.238	0.243	0.248	0.254	0.257	0.270	0.277
	12	0.230	0.234	0.239	0.245	0.247	0.260	0.267
	½	0.461	0.471	0.481	0.491	0.496	0.520	0.534
	¾	0.409	0.418	0.427	0.436	0.441	0.463	0.475
	1	0.366	0.374	0.382	0.390	0.395	0.415	0.427
	1¼	0.333	0.340	0.347	0.355	0.359	0.377	0.387
	1½	0.310	0.316	0.323	0.330	0.334	0.351	0.360
	2	0.286	0.292	0.298	0.304	0.308	0.323	0.331
	2½	0.268	0.274	0.279	0.285	0.289	0.302	0.310
	3	0.252	0.257	0.262	0.268	0.272	0.284	0.292
2	3½	0.241	0.246	0.251	0.257	0.260	0.272	0.280
	4	0.232	0.237	0.242	0.247	0.250	0.262	0.269
	4½	0.225	0.230	0.235	0.240	0.243	0.255	0.262
	5	0.218	0.223	0.228	0.233	0.236	0.247	0.253
	6	0.213	0.217	0.221	0.226	0.228	0.239	0.245
	8	0.200	0.204	0.208	0.213	0.215	0.225	0.231
	9	0.193	0.197	0.201	0.205	0.207	0.218	0.224
	12	0.185	0.190	0.194	0.198	0.200	0.210	0.216

LOW TEMPERATURE PIPE INSULATION

Surfaces maintained at low temperatures should be insulated so as to retard the flow of heat from the outside into the low temperature area and to prevent the formation of condensation and of frost if the temperatures are low enough, as well as to prevent corrosion induced by the presence of condensed moisture on metal surfaces. Materials commonly used for insulating pipes and surfaces at low temperatures are cork, rock cork, hair felt and other felted or fibrous non-absorbent materials. Thermal conductivities of low temperature insulating materials are given in Chapter 5.

Insulating materials are available commercially to meet varying temperature gradients. For example, the thickness of insulation for ice water is approximately 1½-in. if the temperature in the line is not lower than 25 F; the thickness of insulation for brine is approximately 2½ in. where the temperature ranges from 0 deg to 25 F; and the thickness of insulation where the brine temperature ranges from −30 F to zero degrees is approximately 4 in.

Insulation To Prevent Freezing

If the surrounding air temperature remains sufficiently low for an ample period of time, insulation cannot prevent the freezing of still water, or of water flowing at such a velocity that the quantity of heat carried in the water is not sufficient to take care of the heat losses which will result and cause the temperature of the water to be lowered to the freezing point. Insulation can materially prolong the time required for the water to give up its heat, and if the velocity of the water flowing in the pipe is maintained at a sufficiently high rate, freezing may be prevented.

Table 11 may be used for making estimates of the thickness of insulation necessary to take care of still water in pipes at various water and surrounding air temperature conditions. Because of the damage and service interruptions which may result from frozen water in pipes, it is essential that the most efficient insulation be utilized. This table is based on the use of hair felt or cork, having a conductivity of 0.30. The initial water temperature is assumed to be 10 deg above, and the surrounding air temperature 50 deg below the freezing point of water (temperature difference, 60 F).

The last column of Table 11 gives the minimum quantity of water at initial temperature of 42 F which should be supplied every hour for each linear foot of pipe, in order to prevent the temperature of the water from being lowered to the freezing point. The weights given in this column should be multiplied by the total length of the exposed pipe line expressed in feet. As an additional factor of safety, and in order to provide against temporary reductions in flow occasioned by reduced pressure, it is advisable to double the rates of flow listed in the table. It must be emphasized that the flow rates and periods of time designated apply only for the conditions stated. To estimate for other service conditions the following method of procedure may be used.

If water enters the pipe at 52 F instead of 42 F, the time required to cool it to the freezing point will be prolonged to twice that given in the table, or the rate of flow of water may be reduced so that the quantity

Table 11. Data for Estimating Requirements to Prevent Freezing of Water in Pipes

Nominal Pipe Size (Inches)	Number of Hours to Cool Water to Freezing Point			Water Required to Flow to Prevent Freezing Pounds per Linear Foot of Pipe per Hour		
	Thickness of Insulation in Inches					
	1	2	3	1	2	3
½	0.42	0.50	0.57	0.54	0.45	0.40
1	0.83	1.02	1.16	0.68	0.55	0.48
1½	1.40	1.74	2.02	0.84	0.68	0.58
2	1.94	2.48	2.90	0.95	0.75	0.64
3	3.25	4.27	5.08	1.24	0.94	0.79
4	4.55	6.02	7.20	1.47	1.11	0.93
5	5.92	7.96	9.69	1.73	1.29	1.06
6	7.35	9.88	12.20	1.98	1.46	1.19
8	10.05	13.90	17.25	2.46	1.78	1.44
10	13.00	18.10	22.70	2.96	2.12	1.70
12	15.80	22.20	28.10	3.43	2.46	1.93

required will be one-half that shown in the last column of Table 11. However, if the water enters the pipe at 34 F it will be cooled to 32 F in one-fifth of the time given in the table. It will then be necessary to increase the rate of flow so that five times the specified quantity of water will have to be supplied in order to prevent freezing.

If the minimum air temperature is −38 F (temperature difference, 80 F), instead of −18 F, the time required to cool the water to the freezing point will be 60/80 of the time given in the table, or the necessary quantity of water to be supplied will be 80/60 of that given.

In making calculations to arrive at the values given in Table 11, the loss of heat stored in the insulation, the effect of a varying temperature difference due to the cooling of pipe and water, and the resistance of the outer surface of the insulation to the transfer of heat to the air have all been neglected. When these factors enter into the computations it is necessary to enlarge the factor of safety. Also as stated, the time shown in the table is that required to lower the water to the freezing point. A longer period would be required to freeze the water, but the danger point is reached when freezing starts. The flow of water will stop and the entire line will be in danger as soon as the water freezes across the section of the pipe at any point.

When water must remain stationary longer than the times designated in Table 11, the only safe way to insure against freezing is to install a steam or hot water line, or to place an electric resistance heater along the side of the exposed water line. The heating system and the water line are then insulated so that the heat losses from the heating system are not excessive, and the heating effect is concentrated against the water pipe where it is needed. For this form of protection 2 in. of an efficient insulation may be applied.

Pipe Sweating

In some cases the prevention of condensation rather than the conservation of heat is the governing factor in determining the thickness of

insulation required. Fig. 2 may be used for determining the thickness of any material of known conductivity which should be used to prevent condensation on pipes and flat metallic surfaces. The surface resistances used for calculating the family of curves in Fig. 2 are based on the results of tests made on canvas-covered pipe insulation surfaces at Mellon Institute. However, it has been found that the resistance for asphaltic and roofing surfaces is practically the same as for canvas surfaces, so that the curves

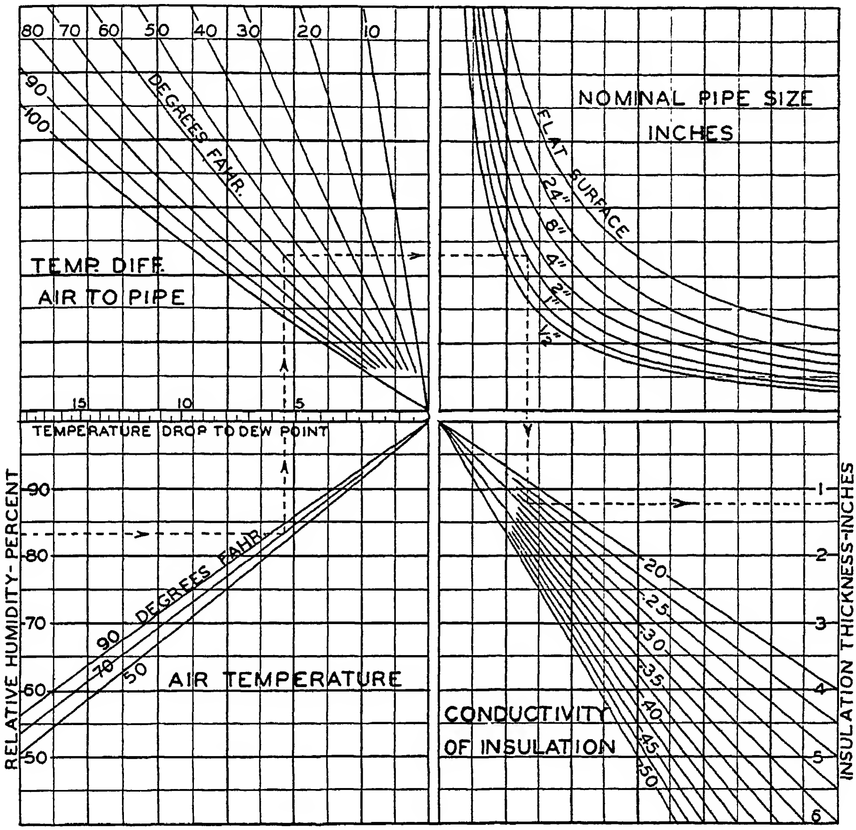
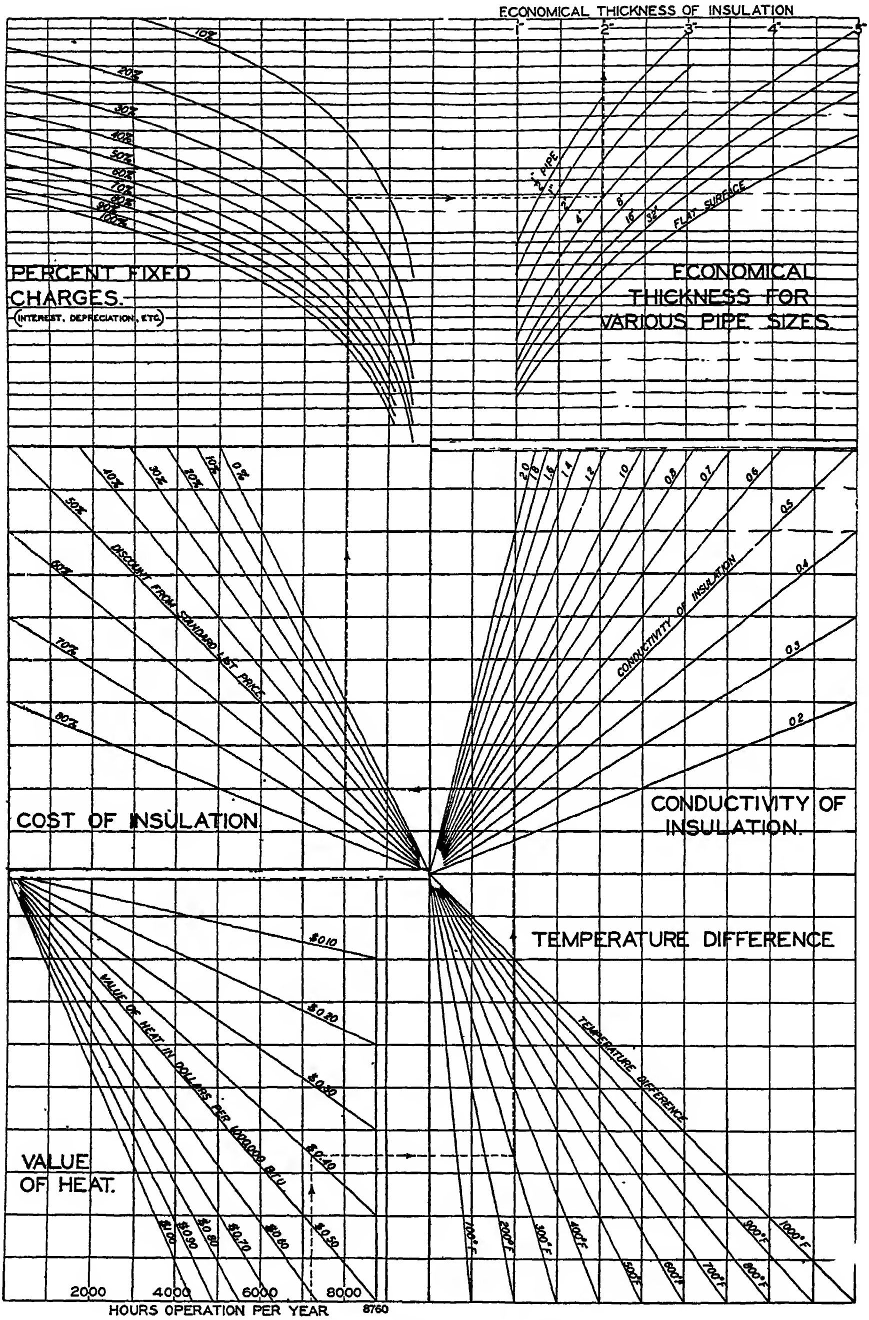


Fig. 2. Thickness of Pipe Insulation to Prevent Sweating[a]

[a]Solve problems by drawing lines as indicated by dotted line, entering chart at lower left hand scale.

given may be followed with no alteration on account of the surfaces commonly used.

Moisture will be deposited on a surface whenever its temperature falls to that of the dew-point. The maximum permissible temperature drop is indicated on Fig. 2 at the point where the guide line passes through the horizontal scale at the left center of the chart. This temperature drop represents the difference between the dry-bulb temperature and the dew-point temperature for the conditions involved. (See discussion of condensation in Chapter 7).



(L. B. McMillan, *Proc. National Dist. Heating Ass'n.*, Vol. 18, p. 131.)

FIG. 3. CHART FOR DETERMINING ECONOMICAL THICKNESS OF INSULATION

The rate of heat loss from a surface maintained at constant temperature is greatly increased by air circulation over the surface. In the case of well-insulated surfaces the increases in losses due to air velocity are very small as compared with increases shown for bare surfaces, because of the fact that air flowing over the surface of the insulation can increase only the rate of heat transfer from surface to air, and cannot change the internal resistance to heat flow inherent in the insulation itself. The maximum increase in loss due to air velocity ranges from about 30 per cent in the case of 1 in. thick insulation, to about 10 per cent in the case of 3 in. thick insulation, provided that the insulation is thoroughly sealed so that air can flow only over the surface.

If the conditions are such that the air may circulate through cracks and crevices in the insulation, the increases may be far greater than those given. Therefore, it is essential that insulation be sealed as tightly as possible. Pipe insulation out-of-doors should be provided with a waterproof jacket, and other outdoor insulation should be thoroughly weather-proofed.

THICKNESS OF PIPE INSULATION

Table 12 shows the thicknesses of insulation which ordinarily are used for various temperature conditions. Where a thorough analysis of economic thickness is desired, this may be accomplished through the use of the chart, Fig. 3.

The dotted line on the chart illustrates its use in solving a typical example. In using the chart, start with the scale at the left bottom margin representing the given number of hours of operation per year; then proceed vertically to the line representing the given value of heat; thence horizontally, to the right, to the line representing the given temperature difference; thence vertically to the line representing the conductivity of the given material; thence horizontally, to the left, to the line representing the given discount on that material; thence vertically to the curve representing the required per cent return on the investment; thence horizontally, to the right, to the curve representing the given pipe size; thence vertically to the scale at the top right margin where the economical thickness may be read off directly. The dotted line on the chart illustrates its use in solving a typical example.

Underground Insulation

Underground steam distribution lines are carried in protective structures of various types, sizes and shapes. (See Chapter 36). Detailed data on commonly used forms of tunnels and conduit systems have been published by the *National District Heating Association*[2].

Pipes in tunnels are covered with sectional insulation to provide maximum thermal efficiency and are also finished with good mechanical protection in the form of metal or waterproofing membrane outer jackets. Conduit systems are in more general use than tunnels. Pipes carried in conduits may be insulated with sectional insulation; however, the more usual practice is to fill the entire section of the conduit around

[2]*Handbook of the National District Heating Association*, Second Edition, 1932.

TABLE 12. THICKNESSES OF INSULATION ORDINARILY USED INDOORS[a]

Steam Pressures (Lb Gage) or Condition	Steam Temperatures Degrees Fahrenheit	Thickness of Insulation		
		Pipes Larger Than 4 In.	Pipes 2 In. to 4 In.	Pipes ½ In. to 1½ In.
0 to 25	212 to 267	1 in.	1 in.	1 in.
25 to 100	267 to 338	1½ in.	1 in.	1 in.
100 to 200	338 to 388	2 in.	1½ in.	1 in.
Low Superheat	388 to 500	2½ in.	2 in.	1½ in.
Medium Superheat	500 to 600	3 in.	2½ in.	2 in.
High Superheat	600 to 700	3½ in.	3 in.	2 in.

[a]All piping located outdoors or exposed to weather is ordinarily insulated to a thickness ½ in. greater than shown in this table, and covered with a waterproof jacket.

the pipes with high quality, loose insulating material. The insulation must be kept dry at all times, and for this purpose effective waterproofing membranes enclose the insulation. A drainage system is also provided to divert water which may tend to enter the conduit.

The economical thickness of insulation for underground work is difficult of accurate determination due to the many variables which have to be considered. As a result of theories developed by J. R. Allen[3], together with experimental data presented by others, the usual endeavor is to secure not less than 90 per cent efficiency for underground piping. Table 13 can be used as a guide in arriving at the minimum thickness of loose insulation fills to use for laying out conduit systems. Other factors such as the number of pipes and their combination of sizes, as well as the standard conduit sizes, are primary controlling factors in the amount and thickness of insulation for use.

When sectional insulation is applied to lines in tunnels or conduits, usual practice is to apply the most efficient materials ½ in. less in thickness than that determined by the use of Fig. 3. Use of Fig. 3 involves conditions of insulation exposed to the air, whereas normal ground temperature is substituted for air temperature in determining the temperature difference for use with the chart when applying it for underground pipe line estimates.

TABLE 13. THICKNESS OF LOOSE INSULATION FOR USE AS FILL IN UNDERGROUND CONDUIT SYSTEMS

Steam Pressure (Lb Gage) or Condition	Steam Temperature Degrees Fahrenheit	Minimum Thickness of Insulation in Inches					Minimum Distance Between Steam and Return
		Steam Lines			Return Lines		
		Pipes Less than 4 In.	Pipes 4 In. to 10 In.	Pipes Larger than 12 In.	Pipes Less than 4 In.	Pipes 4 In. and Larger	
Hot Water, or 0 to 25	212 to 267	1½	2	2½	1¼	1½	1
25 to 125	267 to 352	2	2½	3	1¼	1½	1¼
Above 125, or superheat	352 to 500	2½	3	3½	1¼	1½	1½

[3]Theory of Heat Losses from Pipes Buried in the Ground, by J. R. Allen (A.S.H.V.E. TRANSACTIONS, Vol. 26, 1920).

Chapter 36

DISTRICT HEATING

Underground Steam Piping, Selection of Pipe Sizes, Provision for Expansion, Capacity of Returns with Various Grades, Pipe Conduits, Pipe Tunnels, Service Connections, Steam per Square Foot of Heating Surface, Fluid Meters and Metering

THIS chapter deals with those phases of district heating which frequently fall within the province of the heating engineer. Data and information are included for solving incidental problems in connection with institutions and factories and for the design of building heating systems which are to be supplied with purchased steam. A complete district heating installation should not be attempted without a thorough study of the entire problem by men competent and experienced in that industry. The Handbook and other publications of the *National District Heating Association* and the references at the end of this chapter should be consulted.

UNDERGROUND STEAM PIPING

The methods used in district heating work for the distribution of steam are applicable to any problem involving the supply of steam to a group of buildings. The first step is to establish the route of the pipes, and in this matter the local conditions so fully control the layout that little can be said regarding it.

Having established the route of the pipes, the next step is to calculate the pipe sizes. In district heating work it is common practice to design the piping system on the basis of pressure drop. The initial pressure and the minimum permissible terminal pressure are specified and the pipe sizes are so chosen that the required amount of steam, with suitable allowances for future increases, will be transmitted without exceeding this pressure drop. The steam velocity is therefore almost disregarded and may reach a very high figure. Velocities of 35,000 fpm are not considered high. By the use of this method the pipe sizes are kept to a minimum with consequent savings in investment.

The steam flowing through any section of the piping can be computed from a study of the requirements of the several buildings served. In general a condensation rate of 0.25 lb per hour per square foot of equivalent heating surface is a safe figure. This allows for line condensation which, however, is a small part of the total at times of maximum load. Any unusual requirements such as those for process steam should be individually calculated.

The steam requirements for water heating should be taken into account, but in most types of buildings this load will be relatively small compared

with the heating load and will seldom occur at the time of the heating peak. Unusual features such as large heaters for swimming pools should not be overlooked.

The pressure at which the steam is to be distributed will depend, in part, upon whether or not it has been passed through electrical generating units. If it has, the pressure will be considerably lower than if live steam, direct from the boilers, is used. The advantages of low pressure distribution (2 to 30 lb per square inch) are (1) smaller heat loss from the pipes, (2) less trouble with traps and valves, and (3) simpler problems in pressure reduction at the buildings. With distribution pressures not exceeding 40 lb per square inch there is little danger even if the full distribution pressure should build up in the radiators through the faulty operation of a reducing valve; but with pressures higher than this a second reducing valve or some form of emergency relief is usually desirable to prevent excessive pressures in the radiators. The advantages of high pressure distribution are (1) smaller pipe sizes and (2) greater adaptability of the steam to various operations other than building heating.

The different kinds of apparatus which frequently must be served require various minimum pressures. Kitchen equipment requires from 5 to 15 lb per square inch, the higher pressures being necessary for apparatus in which water is boiled, such as stock kettles and coffee urns. An increased amount of heating surface, which is easily obtained in some kinds of apparatus, results in quicker and more satisfactory operation at low pressures. For laundry equipment, particularly the mangle, a pressure of 75 lb per square inch is usually demanded although 30 lb per square inch is sufficient if the mangle is equipped with a large number of rolls and if a slow rate of operation is permissible. Pressing machines and hospital sterilizers require about 50 lb per square inch.

PIPE SIZES

The lengths of pipe, steam quantities and initial and terminal pressures having been chosen, the pipe sizes can readily be calculated by means of the Unwin pressure drop formula. This formula, which gives pressure drops slightly larger than actual test results, is as follows:

$$P = \frac{0.0001306\, W^2 L \left(1 + \frac{3.6}{d}\right)}{y d^5} \tag{1}$$

where

P = pressure drop, pounds per square inch.
W = weight of steam flowing, pounds per minute.
L = length of pipe, feet.
d = inside diameter of pipe, inches.
y = average density of steam, pounds per cubic foot.

This formula is similar to the Babcock formula given in Chapter 32.

Information on provision for expansion will be found in Chapters 32 and 34.

In general return lines when installed follow the contour of the land, and Table 1 gives sizes of return pipes for various grades. It is evident that at points where the grade is great, smaller pipes can be installed.

PIPE CONDUITS

Conduits for steam pipes buried underground should be reasonably water-proof, able to withstand earth loads and to take care of the expansion and contraction of the piping without strain or stress on the couplings, or without affecting the installation or conduit. Expansion of the piping must be carefully controlled by means of anchors and expansion joints or bends so that the pipes can never come in contact with the conduit. Anchors can be anchor fittings or U-shaped steel straps which partially encircle the pipes and are firmly bolted to a short length of structural steel set in concrete.

TABLE 1. CAPACITY OF RETURNS FOR UNDERGROUND DISTRIBUTION SYSTEMS IN POUNDS OF CONDENSATE PER HOUR

Size[a] of Pipe In.	Pitch of Pipe per 100 Ft.						
	6″	1′	2′	3′	5′	10′	20′
1	448	998	1890	2240	3490	5490	7490
1¼	1740	2490	3990	4880	6480	9480	13500
1½	2700	4190	5740	7480	9480	14500	20900
2	4980	7380	10700	13900	16900	24900	36900
3	13900	22500	30900	37400	50400	74800	105000
4	30900	44800	64800	79700	105000	154000	229000
5	54800	79800	120000	144800	195000	294000	418000
6	90000	138000	187000	237000	312000	449000	....
8	190000	277000	404000	508000	660000	938000	....
10	344000	498000	724000	900000	1190000	....	....
12	555000	798000	1148000	1499000	1990000	....	....

[a]Size of pipe should be increased if it carries any steam.

In laying out conduits of this type the following points should be borne in mind:

1. An expansion joint offset or bend should be placed between each two anchors.
2. If the distance between buildings is 150 ft or less and the steam line contains high-pressure steam, it may be anchored in the basement of one building and allowed to expand into the basement of the second building. If the steam line contains low-pressure steam (up to 4 lb pressure), this method may be used if buildings are 250 ft or less apart.
3. If the distance between buildings is between 150 ft and 300 ft and the steam line contains high-pressure steam, the lines should be anchored midway between the buildings and allowed to expand into the basements of both buildings. If the steam line contains low-pressure steam this method may be used if buildings are between 250 ft and 600 ft apart. No manhole is required at the anchor, and a blind pit is all that is necessary.
4. For longer lines, manholes must be located according to judgment and depending upon the expansion value of the type of expansion joint or bend that is used. The minimum number of manholes will be required when an expansion bend or an anchor with double expansion joint is placed in each manhole and the pipes are anchored midway between manholes.
5. A proper hydrostatic test should be applied to the piping before the top of the conduit is applied and before application of insulation. The pressure used in this test should be greater than the pressure used in service, and should be not less than 100 lb per square inch in any case.

The styles and construction of conduits commonly used may be classified as follows. Some of the more common forms are illustrated in Fig. 1.

Wood Casing: The pipe is enclosed in a cylindrical casing usually having a wall 4 in. thick and built of segments which are bound together by a wire wrapped spirally around

the casing. The casing is lined with bright tin and coated with asphaltum. The pipe is supported on rollers carried in a bracket which fits into the casing. The lengths of casing are tightly fitted together with a male and female joint. This form of conduit is illustrated in Fig. 1 at *A*. The casing rests on a bed of crushed stone with tile drains laid below. The tile drains are of 4-in. field tile or vitrified sewer tile, laid with open joints.

Filler Type: The pipes are supported on expansion rollers properly supported from the conduit or independent masonry base. The pipes are protected by a split-tile conduit, and the entire space between the pipes and the tile is filled with an insulating filler. Thus the pipes are nested and the insulation between them and the tile effectively prevents circulation of air. The conduit is placed on a bed of gravel or crushed rock from 4 to 6 in. thick, which is extended upward so as to come about 2 in. above the parting lines of the tile. A tile underdrain is placed beneath the conduit throughout the entire length and is connected to sewers or to some other point of free discharge. At *B* and *D* in Fig. 1 are shown two forms of tile conduit of the filler type.

Circular Tile or Cast-Iron Conduit: The pipes are carried on expansion rollers supported on a frame which rests entirely on the side shoulders of the base drain foundation.

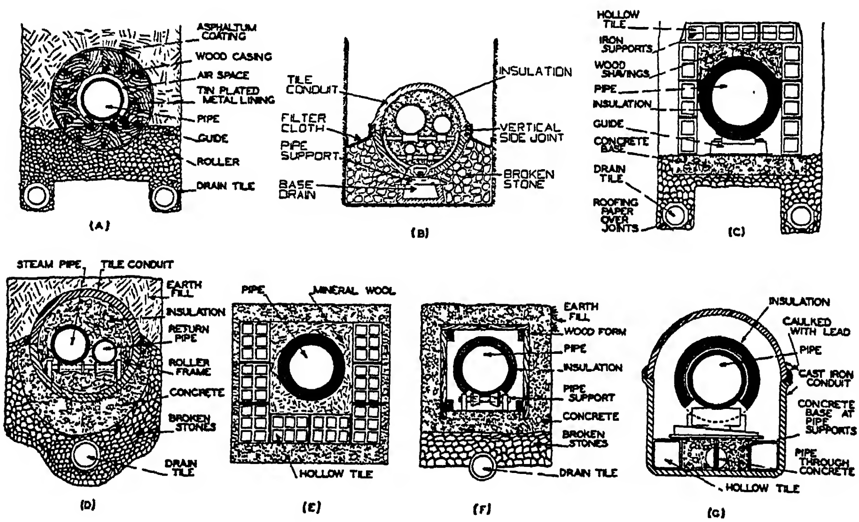


FIG. 1. CONSTRUCTION DETAILS OF CONDUITS COMMONLY USED

The pipes are protected by a sectional tile conduit, scored for splitting, or a cast-iron conduit, both being of the bell and spigot type. The conduit has a longitudinal side joint for cementing, after the upper half of conduit is in place, so shaped that the cement is keyed in place while locking the top and bottom half of the conduit together with a water-tight vertical side joint. The cast-iron conduit has special side locking clamps in addition to the vertical side joint. The entire space between the conduit and the pipes is filled with a water-proofed asbestos insulation. The conduit is supported on the base drain foundation, each section resting on two sections of the base drain, thus interlocking. The base drain is so shaped that it provides a cradle for the conduit, resting solidly on the trench bottom and providing adequate drainage area immediately under the conduit. The underdrain is connected to sewers or some other point of free discharge. For tile conduit the base drain is vitrified salt glazed tile and for cast-iron conduit it is either extra heavy tile or cast-iron. A free internal drainage area is also provided to carry away any water that may collect on the inside of the conduit from a leaky pipe or joint in the conduit. Broken stone is filled in around the base drain and up to the vertical side joint. The broken stone is covered with an asphalted filter cloth to prevent sand, etc., from sifting through the broken stone and clogging the drainage area of the base drain. The tile conduit is made in 2 ft lengths and the cast-iron conduit in 4 ft lengths, cast in

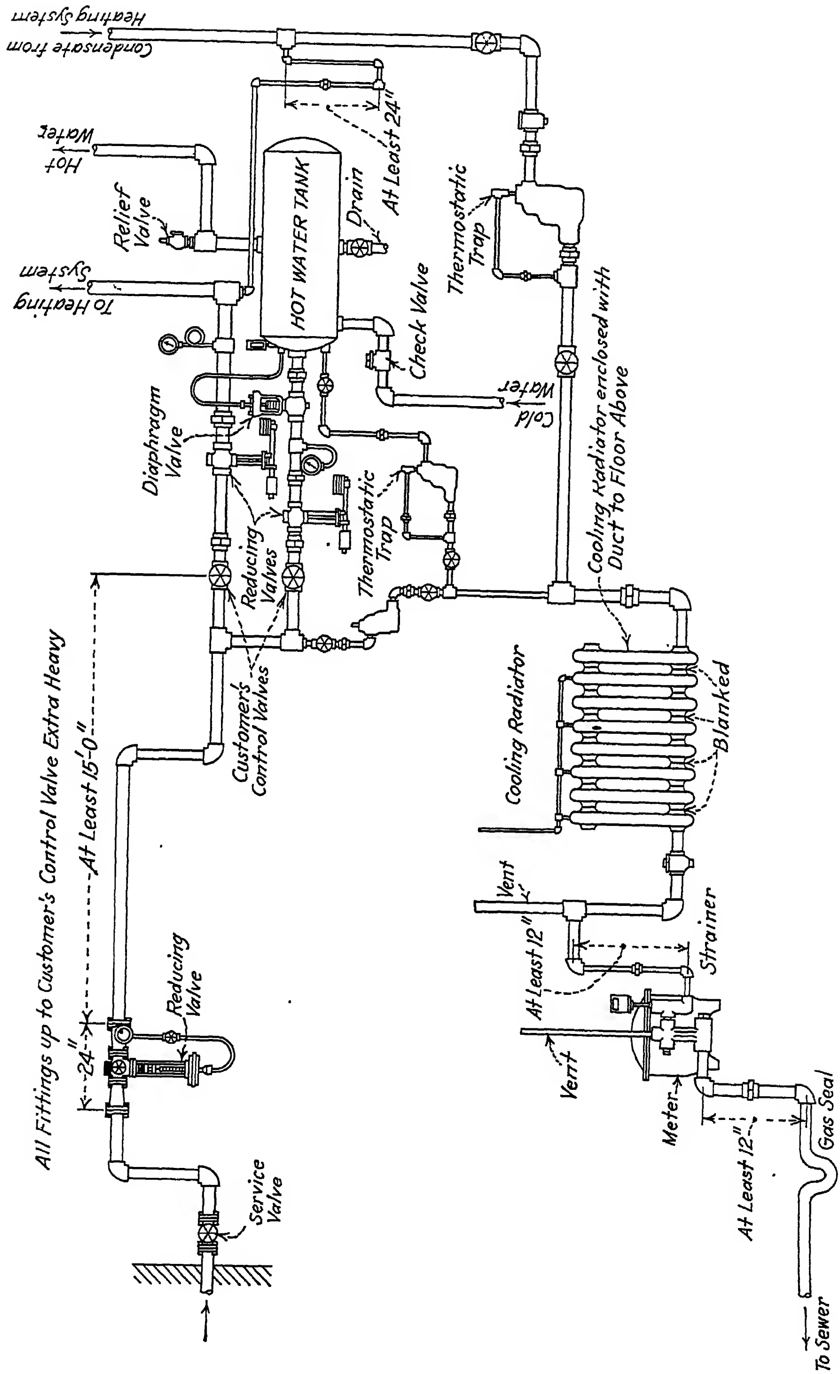


FIG. 2. STEAM SERVICE AND METER CONNECTIONS WITH COOLING RADIATOR

separate top and bottom halves. Special reinforcing ribs give the cast-iron conduit ample strength with minimum weight.

Insulated Tile Type: The insulating material, diatomaceous earth, is molded to the inside of the sectional tile conduit. The space between the pipes and the insulating conduit lining may also be filled with insulation. The pipes are carried on expansion rollers supported on a frame which rests on the side shoulders of the base drain foundation. This type of conduit has the same mechanical features as described under the heading Circular Tile or Cast-Iron Conduit.

Sectional Insulation Type (Tile or Cast-Iron): Each pipe is insulated in the usual way with any desired type of sectional pipe insulation over which is placed a standard waterproof jacket with cemented joints. The pipes are enclosed in a sectional tile or cast-iron conduit as described under the heading Circular Tile or Cast-Iron Conduits.

Sectional Insulation Type (Tile or Concrete Trench): A type of construction frequently used in city streets, where service connections are required at frequent intervals, the pipes are insulated as described in the preceding paragraph, and are enclosed in a box or trench made either entirely of concrete, or with concrete bottom and specially constructed tile sides and tops. The pipes are supported on roller frames secured in the concrete. At *C* and *E*, Fig. 1, are shown two tile conduits using sectional insulation. In these particular designs the space surrounding the pipe is filled partially or wholly with a loose insulating material. The use of loose material in addition to the sectional insulation is, of course, optional and is only justifiable where high pressure steam is used. The conduit shown at *F* is of a similar type and has the advantage of being made entirely of concrete and other common materials.

Sectional Insulation Type (Bituminized Fibre Conduit): Each pipe is individually insulated and encased in a bituminized fibre conduit. The insulating material is 85 per cent carbonate of magnesia sectional pipe covering, applied in the usual manner as on overhead pipes, except that bands are omitted. After every fifth section of magnesia covering there is applied a short, hollow section of very hard asbestos material in the bottom portion of which rests a grooved-iron plate carrying ball-bearings upon which the pipe rides when expanding or contracting. This short expansion section is of the same outside diameter as the adjacent 85 per cent magnesia covering. Over the pipe covering and expansion device there are placed two layers of bituminized fibre conduit with all joints staggered, and the surface of each conduit is finished with liquid cement. Conduits are placed on a bed of crushed rock or gravel, approximately 6 in. deep, and this is extended upward to about the center line of the conduit when trench is backfilled. Underdrains leading to points of free discharge are placed in the gravel or crushed rock beds.

Special Water Tight Designs: It is occasionally necessary to install pipes in a very wet ground, which calls for special construction. The ordinary tile or concrete conduit is not absolutely water tight even when laid with the utmost care. The conduit shown at *G*, Fig. 1, is of cast-iron with lead-calked joints and is water tight if properly laid. It is obviously expensive and is justified only in exceptional cases. A reasonably satisfactory construction in wet ground is the concrete or tile conduit with a waterproof jacket enclosing the pipe and its insulation, and with the interior of the conduit carefully drained to a manhole or sump having an automatic pump. It is useless to install external drain tile when the conduit is actually submerged.

PIPE TUNNELS

Where steam heating lines are installed in tunnels large enough to provide walking space, the pipes are supported by means of hangers or roller frames on brackets or frame racks at the side or sides of the tunnel. The pipes are insulated with sectional pipe insulation over which is placed a sewed-on, painted canvas jacket or a jacket of asphalt saturated asbestos water-proofing felt. The tunnel itself is usually built of concrete or brick and water-proofed on the outside with membrane water-proofing.

On account of their relatively high first cost as compared with smaller conduits, walking tunnels are sometimes not installed where provision for the heating lines is the only consideration, but only where they are required

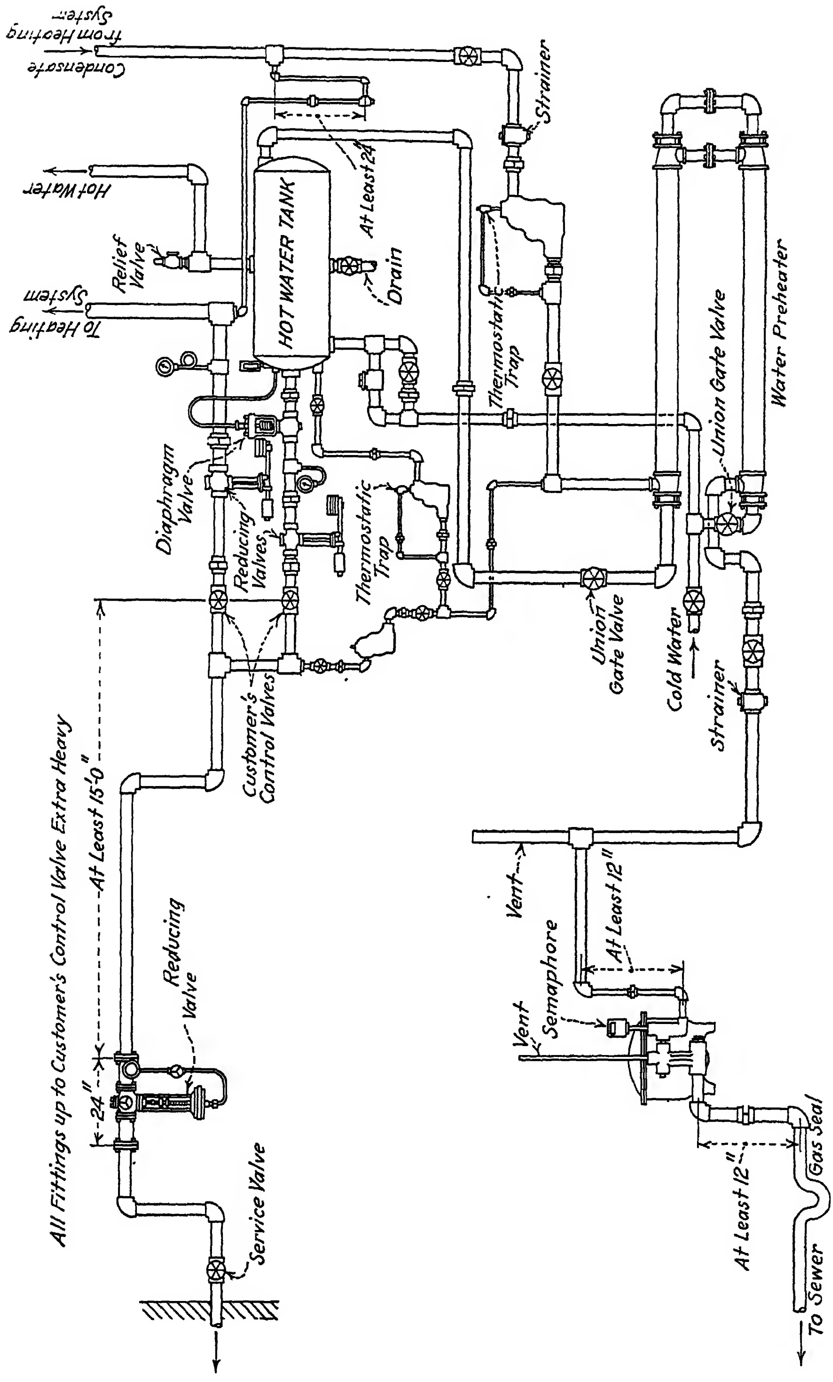


FIG. 3. STEAM SERVICE AND METER CONNECTIONS WITH WATER PREHEATER

to accommodate miscellaneous other services, such as for underground passage between buildings.

SERVICE CONNECTIONS

Most district heating companies enforce certain regulations regarding the consumer's installation, partly to safeguard their own interests but principally to insure satisfactory and economical service to the consumer. There are certain fundamental principles that should be followed in the design of a building heating system which is to be supplied from street mains. Although some of these apply to any building, they have been demonstrated to be especially important when steam is purchased.

1. Provision should be made for conveniently shutting off the steam supply at night and at other times when heat is not needed.

It has been thoroughly demonstrated that a considerable amount of heat can be saved by shutting off steam at night. Although there is, in some cases, an increased consumption of heat when steam is again turned on in the morning, there is a large net saving which may be explained by the fact that the lower inside temperature maintained during the night obviously results in lower heat loss from the building, and less heat need therefore be supplied.

Steam can be entirely shut off at night in most buildings even in very cold weather without endangering plumbing. It is necessary, however, to have an ample amount of heating surface so that the building can be quickly warmed in the morning. Where the hours of occupancy differ in various parts of the building, it is good practice to install separate supply pipes to the different parts. For example, in an office building with stores or a restaurant on the first floor which are open in the evening, a separate main supplying the first floor will permit the steam to be shut off from the remainder of the building in the late afternoon. The division of the building into zones each with a separately controlled heat supply is sometimes desirable, as it permits the heat to be adjusted according to variations in sunshine and wind.

2. Residual heat in the condensation should not be wasted.

This heat may be salvaged by means of a cooling radiator as illustrated in Fig. 2, or, as is more frequently done, by a heat exchanger which preheats the water used for lavatory purposes, as in Fig. 3. Figs. 2 and 3 show the practice of a Boston utility as to the service and meter connections. Fig. 4 shows a typical installation for service from the system of a New York utility.

The condensation from the heating system, after leaving the trap, passes through the preheater on its way to the meter. The supply to the hot water heater passes through the preheater, absorbing heat from the condensation. If the hot water system in the building is of the recirculating type, the recirculating connection should be tied in *between* the preheater and the water heater proper, not at the preheater inlet, because the recirculated hot water is itself at a high temperature. The number of square feet of heating surface in the preheater should be approximately equal to one per cent of the equivalent square feet of heating surface in the building.

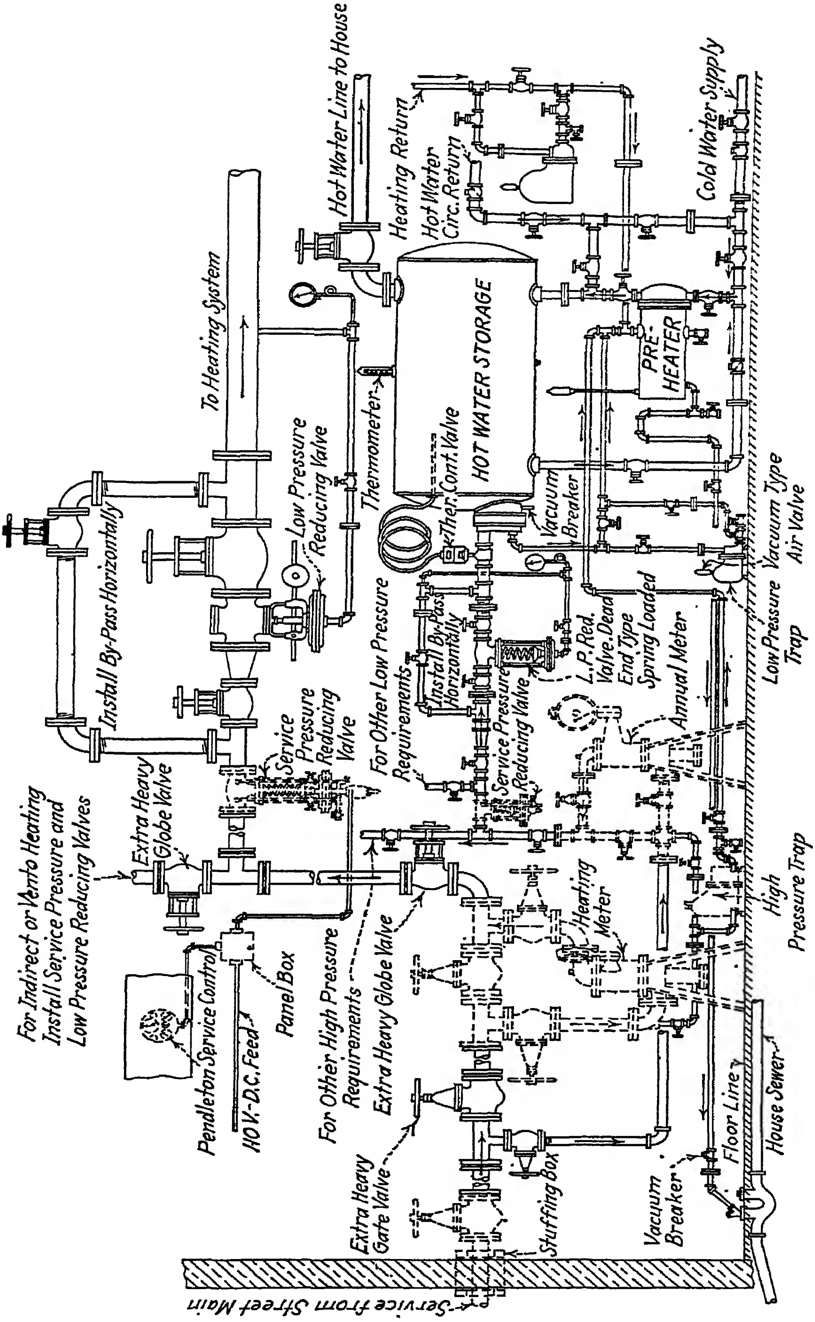


FIG. 4. TYPICAL SERVICE INSTALLATION

Because of the lack of coincidence between the heating system load and the hot water demand, a greater amount of heat can be extracted from the condensation if storage capacity is provided for the preheated water. Frequently a type of preheater is used in which the coils are submerged in a storage tank.

3. Heat supply should be graduated according to variations in the outside temperature.

This may be done in several ways, as by the use of thermostats of various types or by orifice systems. Another method which is very simple is the use of an ordinary vacuum return line system in which the pressure in the radiators is varied between a high vacuum and a few pounds pressure, thus producing some control over the heat output. One form of control which appears to be well suited for controlling district steam service to a building is the weather compensating thermostat. It regulates the steam supply automatically according to the outdoor temperature, and gives frequent short intervals of intermittent steam supply, and at the same time insures delivery of steam to all the radiators. Another form of regulation, known as the time-limit control, is sometimes employed for regulating the steam supply from the central station main to the building.

Such a control provides an intermittent supply of steam to the radiation either throughout the 24 hrs of the day or during the daytime hours only. The setting of a switch may provide no service, continuous service, or periodic service. For the latter, by means of several intermittent settings, steam will be supplied during each period in increments of a certain number of minutes for each successive setting of the switch, steam being shut off during the balance of the period. These settings afford from 15 to 80 per cent of the maximum heating effect required on days of zero temperature.

A night switch with a variety of settings may be adjusted so as to maintain throughout the night the intermittent supply called for by the day switch setting, or may be set to interrupt the operation of the day switch and entirely cut off the supply of steam to the radiation at night during certain hours which are selected by the operating engineer.

FLUID METERS

No one thing has contributed more to the advancement of district heating than that of the perfection of fluid meters, which may be classified as follows:

1. *Positive Meters:* The fluid passes in successive isolated quantities—either weights or volumes. These quantities are separated from the stream and isolated by alternately filling and emptying containers of known capacity.

2. *Differential Meters:* The fluid does not pass in isolated separately-counted quantities but in a continuous stream which may flow through the line without actuating the primary device of the meter. In the differential meter, the quantity of flow is not determined by simple counting, as with the positive meter, but is determined from the action of the steam on the primary element.

Additional subdivisions of these two general classifications can be made as follows:

- Fluid Meters
  - Positive - quantity
    - Weighing
      - Weighers
      - Tilting trap
    - Volumetric
      - Rotary
      - Bellows
  - Differential
    - Quantity - Current - Turbine
    - Rate of flow
      - Head (Kinetic)
        - Venturi
        - Flow nozzle
        - Orifice
        - Pitot tube
      - Area (Geometric)
        - Orifice and plug
        - Cylinder and piston
      - Head area (Weir)
        - V-notch
        - Special notch

In selecting a meter for a particular installation, the number of different makes and types of meters suitable for the job is usually limited by one or more of the following considerations:

1. Its use in a new or an old installation.
2. Method to be used in charging for the service.
3. Location of the meter.
4. Large or small quantity to be measured.
5. Temporary or permanent installation.
6. Cleanliness of the fluid to be measured.
7. Temperature of the fluid to be measured.
8. Accuracy expected.
9. Nature of flow: turbulent, pulsating or steady.
10. Cost.
    (*a*) Purchase price.
    (*b*) Installation cost.
    (*c*) Calibration cost.
    (*d*) Maintenance cost.
11. Servicing facilities of the manufacturer.
12. Pressure at which fluid is to be metered.
13. Type of record desired as to indicating, recording or totalizing.
14. Stocking of repair parts.
15. Use of open jets where steam is to be metered.
16. Metering to be done by one meter or by a combination of meters.
17. Use as a check meter.
18. Its facilities for determining or recording information other than flow.

Condensation Meters

The majority of the meters used by district heating companies in the sale of steam to their customers are of the condensation or flow types.

The condensation meter is a popular type for use on small and medium sized installations, where all of the condensate can be brought to a common point for metering purposes. Its simplicity of design, ease in testing, accuracy at all loads, low cost, and adaptability to low pressure distribution has made it standard equipment with many heating companies.

Two types of condensation meters are in general use: the *tilting bucket* meter and the *revolving drum* or *rotor* meter of which there are several makes on the market.

Condensation meters should not be operated under pressure. They are made for either gravity or vacuum installation. Continuous flow traps and vented receivers ahead of gravity type meters are desirable. A typical

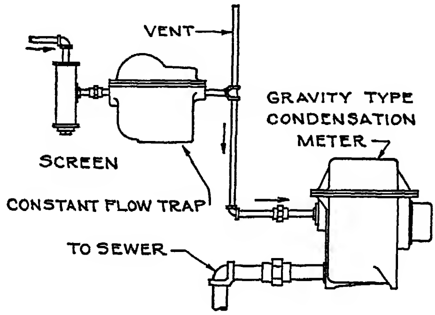


FIG. 5. TYPICAL GRAVITY INSTALLATION OF CONDENSATION METER

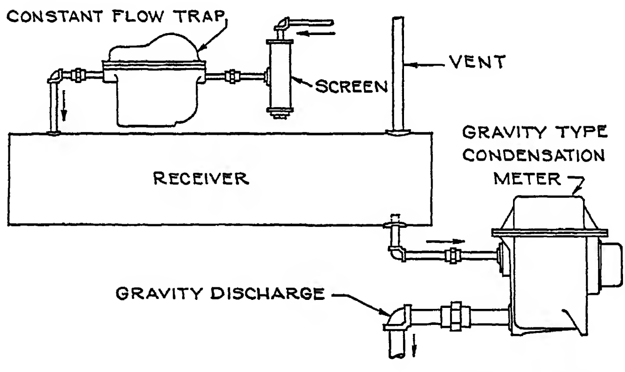


FIG. 6. GRAVITY INSTALLATION FOR CONDENSATION METER USING VENTED RECEIVERS

gravity installation for a condensation meter is shown in Fig. 5, while Fig. 6 is a gravity installation using a vented receiver ahead of the meter. Fig. 7 shows a vacuum installation without a master trap. The pipe discharge from the meter should be of ample size pitched so as to rapidly remove the condensate from the meter. It is advisable to follow the instructions of the meter manufacturer in installing a meter as the successful operation of any meter depends upon its being properly connected.

Steam flow meters are available in many types and combinations, as indicated in the subdivision covering fluid meters on page 528.

The *orifice and plug* meter is one in which the steam flow varies directly

as the area of the orifice. The vertical lift of the plug, which is proportional to the flow, is transmitted by means of a lever to an indicator and to a pencil arm which records the flow on a strip chart. The total flow over a given period is obtained by measuring the area by using a planimeter on the chart and applying the meter constant.

Flow meters using an orifice, Venturi tube, flow nozzle, or Pitot tube as the primary device are made by a number of manufacturers and can be obtained in either the mechanical or electrically operated type. The electric flow meter makes it possible to locate the instruments at some

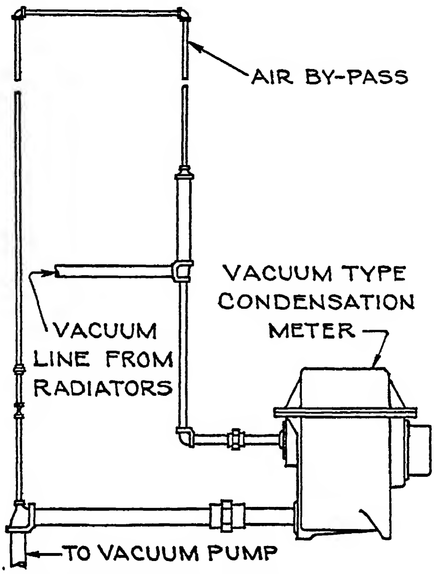


Fig. 7. Vacuum Condensation Meter Installation without Master Trap

distance from the primary element, which is not always possible with the mechanical flow meter.

Flow meters employing the orifice, Venturi tube, flow nozzle or Pitot tube should be so selected as to keep the lower operating range of the load above 20 per cent of the capacity of the meter. This is desirable for accuracy as the differential pressure at light loads is too small to properly actuate the meter. A few general points to be considered in installing a meter of this type are as follows:

1. It is desirable to place the differential medium in a horizontal pipe in preference to a vertical one, where either location is available.

2. Reservoirs should always be on the same level and installed in accordance with the instructions of the meter company.

3. The meter body should be placed at a lower level than that of the pressure differential medium. Special instructions are furnished where the meter body is above.

4. Meter piping should be kept free from leaks.

5. Sludge should not be permitted to collect in the meter body.

6. The meter body and meter piping should be kept from freezing temperatures.
7. It is best not to connect a meter body to more than one service.
8. Special instructions are furnished for metering a turbulent or pulsating flow.

STEAM PER SQUARE FOOT OF HEATING SURFACE

The following factors are used in New York City for the different classes of buildings listed. The factors are based on maintaining an inside temperature of 70 F for certain hours with a minimum outside temperature of 0 deg F, and an average of 43 F for the heating season of eight months (October 1 to June 1). In this group are six types of buildings:

For manufacturing or commercial loft type where steam is used to heat the premises during the day hours to maintain 65 to 68 F from 9 a.m. to 5 p.m. No Sunday or holiday use and no night use. *Factor:* 325 lb per square foot of heating surface per season.

For office buildings using steam during daylight hours only to maintain 70 F from 9 a.m. to 6 p.m. for approximately 240 days (heating season). No night use. *Factor:* 400 lb per square foot of heating surface per season.

For office buildings using steam during day hours and at night when required to 7, 8 and 9 p.m. (customary where there are stock brokers or banking offices), 240 days. *Factor:* 500 lb per square foot of heating surface per season.

For residences of the block type (not detached) where high-class heating service is required somewhat similar to apartment buildings. *Factor:* 550 lb per square foot of heating surface per season.

For apartment houses where high-class heating service is required. (Steam off at midnight). *Factor:* 650 lb per square foot of heating surface per season.

For hotels (commercial type) where very high-class service is required for 24 hours. *Factor:* 800 lb per square foot of heating surface per season.

By assuming one square foot of equivalent heating surface for each 100 cu ft of space heated, which seems a fair ratio in New York City, it is possible roughly to estimate the steam required per cubic foot of space, information which is often more easily obtained than the square feet of heating surface. Additional data on the heating requirements of various types of buildings in a number of cities may be found in the Handbook of the *National District Heating Association.*

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Chapter 37

RADIANT HEATING

Physical and Physiological Considerations, British Equivalent Temperature, Control of Heat Losses, Methods of Application, Principles of Calculation, Mean Radiant Temperature, Measurement of Radiant Heating

THE general theory behind heating for comfort is that heat must be supplied to regulate the rate of heat loss from the human body so that the physiological reactions are conducive to a feeling of comfort in the individual. While in convection heating, as described in Chapter 30, heat is transferred from a heating unit to the air and thence to the occupant, the primary object of radiant heating is to warm the occupant directly without heating the air to any extent. Thus, the difference between convection heating and radiant heating is partly physical and partly physiological.

PHYSICAL AND PHYSIOLOGICAL CONSIDERATIONS

Comfort requires that heat be removed from the body at the same rate as it is generated by the oxidation of the foodstuffs in the body tissues. The normal rate of heat production in a sedentary individual is about 400 Btu per hour[1], or, since the entire surface area of an average adult is 19.5 sq ft, about 20.5 Btu per square foot per hour. Conditions should be such as to remove heat at this rate if the surface is to be maintained at the mean normal surface temperature of the human body.

Heat is transferred from any warm, dry body to cooler surroundings principally by convection and by radiation, the total rate of heat loss being the sum of the two. Where the body surface is moist there is additional loss of heat through evaporation from both the body surface and the respiratory tract.

The rate of heat loss by convection depends upon the difference between the temperature of the body and that of the surrounding air, and on the rate of air motion over the body. The loss by radiation depends entirely upon the difference between the temperature of the body and the mean surface temperature of the surrounding walls and objects. This latter difference is called the *mean radiant temperature* (MRT). Because these two types of heat loss act in a supplementary manner toward each other, a required rate of heat loss can be secured by having a relatively low air temperature and a relatively high MRT, or *vice versa*. Thus, if the air is

[1]Heat and Moisture Losses from the Human Body and Their Relation to Air Conditioning Poblems, by F. C. Houghten, W. W. Teague, W. E. Miller, and W. P. Yant (A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929).

reduced from a given temperature to a lower temperature, the amount of heat lost from the body by convection is increased, and this increase can be compensated for by raising the MRT. Similarly, with a higher air temperature the same total heat loss will be maintained by a correspondingly lower MRT.

The loss by evaporation depends on the air temperature, air movement, and humidity; it is increased if the humidity is reduced. For the usual conditions of heating by radiators or convectors, where the air temperature ranges from 70 F to 73 F, approximately 75 per cent of the total heat loss of 400 Btu per hour occurs by radiation and convection, and the balance, or 100 Btu per hour, occurs by evaporation. In the case of radiant heating, if the air temperature is reduced to 60 F, 84 per cent of the 400 Btu per hour, or 336 Btu per hour, is lost by radiation and convection, and 64 Btu per hour are lost by evaporation.

The mean normal surface temperature of the human body, taken over the whole area, including not only the exposed skin surface but also surfaces of the clothes and the hair, has been very extensively used as 75 F, particularly in British literature. However, results obtained by Aldrich[2] in rooms in which the air and wall surface temperatures were approximately 72 F gave mean values nearer to 83 F than 75 F.

The mean body surface temperature which will maintain the optimum heat loss by radiation and convection in a uniform environment of 72 F may be calculated from fundamental equations for radiation and natural convection by substituting a comparable cylinder for the body. Heilman[3] gives the following equations:

$$H_r = 0.1723\, e \left[\left(\frac{T_s}{100}\right)^4 - \left(\frac{T_w}{100}\right)^4 \right] \tag{1}$$

$$H_c = 1.235 \left(\frac{1}{D}\right)^{0.2} \times \left(\frac{1}{T_m}\right)^{0.181} \times \left(T_s - T_a\right)^{1.266} \tag{2}$$

where

H_r = heat loss by radiation in Btu per square foot per hour.
H_c = heat loss by convection in Btu per square foot per hour.
T_s = absolute temperature of the body surface in degrees Fahrenheit.
T_w = absolute temperature of the walls in degrees Fahrenheit.
T_a = absolute temperature of the air in degrees Fahrenheit.
$T_m = \dfrac{T_s + T_a}{2}$
D = diameter of cylinder in inches.
e = the ratio of actual emission to black body emission.

By assuming for a normal adult an average height of 5 ft 8 in. and an average body surface of 19.5 sq ft, an equivalent diameter of 13.15 in. is obtained. The value of e for skin and clothing is practically 0.95. If both T_a and T_w are taken as 72 F, or 532 absolute, and the sum of H_r and H_c is made 15.4 Btu per square foot per hour, solution of these equations gives a value of approximately 83 F for the normal temperature

[2]A study of Body Radiation, by L. B. Aldrich (Smithsonian Miscellaneous Collections, Vol. 81, No. 6, December, 1928).

[3]Surface Heat Transmission, by R. H. Heilman (*Trans. A.S.M.E., Fuels and Steam Power Section*, Vol. 51, No. 22, September-December, 1929).

of the body surface. This agrees more closely with the values obtained by Aldrich than with the 75 F used by British investigators.

British Equivalent Temperature

The British equivalent temperature (BET) is an index number used in radiant heating considerations which indicates the rate of heat loss, by radiation and convection only, from a body in still air maintained at a surface temperature of 83 F. As originally defined, this index was based on a surface temperature of 75 F, but 83 F has been accepted as giving results more nearly conforming with American practice[4]. When the mean radiant temperature is the same as the air temperature, this value is also that of the BET, but when there is a difference between the two, the BET is always intermediate. The higher the BET, the less the heat loss from the body, the rate of loss in still air being approximately proportional to the difference between the BET and the mean body surface temperature.

If the BET were 83 F, there could be no loss of heat from a surface at that temperature, so the temperature of a normal body surface would have to rise to a point where the heat generated in the tissues could be dissipated.

When convected heat is used, the temperatures of the air and walls are nearly the same, and the optimum value of the BET from the physiological point of view is 72 deg Fahr. Under these conditions the mean surface temperature of a normal body would have the optimum value of 83 F because the rate of heat loss by radiation and convection would be 15.4 Btu per square foot per hour and that by evaporation, 5.1 Btu per square foot per hour, which would just balance the rate of heat production of 20.5 Btu per square foot per hour. This BET of 72 deg Fahr in a uniform environment is exactly equivalent to the *effective temperature* of 66 deg Fahr as defined by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS (see Chapter 2), because, in a uniform environment, a dry bulb temperature of 72 F in still air with a relative humidity of 30 per cent gives an *effective temperature* of 66 deg Fahr, which has been determined to be the optimum.

In radiant heating, where the air may differ considerably in temperature from the surrounding objects, a change occurs in the relations among the heat lost by radiation, convection, and evaporation. It would seem, therefore, that a modification should be made in the optimum BET. If the air temperature drops as low as 60 F the heat loss by evaporation is reduced to 64 Btu per hour, and that by radiation and convection must be increased to 336 Btu per hour in order to maintain the total loss of 400 Btu per hour needed for optimum comfort. This 336 Btu per hour, or 17.2 Btu per square foot per hour, corresponds to a BET of 71 deg Fahr. Hence, for all practical purposes, the optimum BET of 72 deg Fahr may be regarded as applicable to both convection heating and radiant heating.

METHODS OF APPLICATION

There are two general methods of application of radiant heating, as follow:

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[4]Application of the Eupatheoscope for Measuring the Performance of Direct Radiators and Convectors in Terms of Equivalent Temperatures, by A. C. Willard, A. P. Kratz, and M. K. Fahnestock (A.S.H.V.E. Journal, *Heating, Piping and Air Conditioning*, July, 1933).

1. *By warming the interior surfaces of the building.* Pipe coils are embedded in the concrete or plaster of the walls, ceiling or floors, the heating medium being hot water or, in some cases, steam. The temperature of the heating medium should not exceed about 120 F on account of the possibility of cracking the plaster. The area of the panel can be sufficient to supply the requisite quantity of heat at this low temperature. This has the effect of warming the entire concrete or plaster surface in which the pipes are embedded. When carefully designed, this method produces comfortable and economical results.

2. *By separate heated plates or panels attached to the interior surfaces of the structure.* These plates or panels are placed either in an insulated recess and flush with the surface of the walls or ceiling or bolted on its face, and may be decorated as desired. As it is difficult to make an invisible joint between the edge of such a plate and the plaster, it is common to use a frame, either of plaster, wood, metal or composition, around the panel itself. These plates may be placed either on the ceiling or the wall, or in some cases as a margin around the edge of the floor. If floor heating is required the temperature over the whole area should not exceed 70 F. They may be erected conveniently in the form of a dado along the whole length of a long room, as for example, a hospital ward, or they may be designed to form part of the panelling under a window, or in other positions. They are capable of a wide variety of applications.

If the entire warm surface is installed at one end of the room there may be a marked difference between the BET on the two sides of a body in the room. It is usually desirable therefore that the heat be distributed at different points in the room so that no uncomfortable effects will be felt from unequal heating.

PRINCIPLES OF CALCULATION

The calculations for radiant heating are entirely different from those for convective heating. The purpose of the latter is to determine the rate of heat loss from the room by conduction, convection and radiation when maintained in the desired condition; radiant heating involves the regulation of the rate of heat loss per square foot from the human body.

The first step in the calculations for radiant heating is to ascertain the necessary mean radiant temperature (MRT); next, the size, temperature and disposition of the heating surfaces required in the room to produce this MRT are estimated; and after this the determination of the convective heat is made.

Mean Radiant Temperature

If the whole of the interior surface of a room were at the same temperature, this temperature would represent the MRT. Such a condition seldom exists, however, since the actual surface temperature in any heated space having surfaces exposed to the outer air varies greatly for different sides of the enclosure. It is therefore necessary to ascertain by calculation the mean of these interior surface temperatures.

The mean temperature in this sense is not the arithmetic average of the actual thermometric temperatures of the surfaces, but the temperature corresponding to the average rate of heat emission per square foot of surface. The temperature corresponding to this mean emission can be taken from Table 1. Conversely, the emission at different temperatures and emissivity factors can be obtained from this table. For instance, 1 sq ft of surface at 50 F will emit 104.9 Btu per square foot per hour to surroundings at absolute zero if the emissivity of the surface is 0.9.

If the area in square feet of each part of the space is multiplied by the emission value corresponding to its actual temperature and these products

are added together, the gross total amount of radiant heat discharged into the room by the wall surface per hour is obtained. This quantity, divided by the total interior surface, gives the average amount of heat coming into the room from the surface of the walls per square foot of surface per hour.

According to Table 1, the total radiation from a surface at 83 F for an emissivity of 0.95 is 142 Btu per square foot per hour. The difference

TABLE 1. TOTAL BLACK BODY RADIATION TO SURROUNDINGS OF ABSOLUTE ZERO[a]

Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e				Body or Mean Radiant Temperature Deg Fahr	Radiation in Btu per square foot per hour emitted to surroundings with a temperature of absolute zero by bodies at various temperatures and with emissivity factor e			
	e 1.00	e 0.95	e 0.90	e 0.80		e 1.00	e 0.95	e 0.90	e 0.80
30	99.3	94.3	89.4	79.4	71	136.5	129.6	122.9	109.3
35	103.5	98.3	93.2	82.8	72	137.4	130.5	123.6	109.9
40	107.6	102.4	96.8	86.1	73	138.4	131.5	124.5	110.6
45	112.1	106.5	100.9	89.7	74	139.6	132.6	125.6	111.7
46	112.9	107.3	101.6	90.4	75	141.0	133.9	126.9	112.8
47	113.9	108.2	102.5	91.1	80	146.6	139.4	132.0	117.4
48	114.8	109.1	103.4	91.9	85	152.3	144.6	137.1	121.9
49	115.6	109.9	104.1	92.4	90	157.9	149.9	142.1	126.4
50	116.5	110.6	104.9	93.2	100	169.6	161.1	152.6	135.7
51	117.5	111.6	105.8	94.0	110	181.6	172.5	163.5	145.4
52	118.4	112.5	106.5	94.7	120	194.8	185.0	175.4	155.9
53	119.4	113.4	107.4	95.5	130	210.1	199.6	189.1	168.1
54	120.2	114.2	108.2	96.2	140	223.2	212.1	201.0	178.5
55	121.1	115.1	109.0	96.9	150	237.1	225.2	213.5	189.7
56	122.1	116.0	109.9	97.7	160	251.1	238.8	226.0	201.0
57	123.1	117.0	110.9	98.5	170	270.5	257.0	243.5	216.4
58	124.0	117.8	111.6	99.2	180	288.0	273.8	259.1	230.4
59	124.9	118.6	112.4	99.9	190	306.5	291.0	275.8	245.1
60	125.8	119.5	113.4	100.7	200	325.2	309.0	292.8	260.3
61	126.6	120.3	114.0	101.4	210	348.0	330.6	313.1	278.4
62	127.7	121.4	114.9	102.2	220	371.5	353.0	334.4	297.1
63	128.6	122.2	115.8	102.9	250	437.8	415.9	394.0	350.2
64	129.6	123.1	116.7	103.7	300	575.0	546.1	517.5	460.0
65	130.5	124.0	117.5	104.4	350	740.0	703.0	666.0	592.0
66	131.6	125.0	118.4	105.4	400	942.1	895.0	847.5	753.5
67	132.5	125.9	119.3	106.0	450	1176.0	1117.0	1059.0	941.0
68	133.5	126.8	120.1	106.8	500	1464.0	1390.0	1318.0	1171.0
69	134.5	127.8	121.1	107.6	550	1791.0	1701.0	1613.0	1434.0
70	135.5	128.8	121.9	108.4	600	2405.0	2284.0	2165.0	1925.0

[a]These factors are calculated from the formula:

$$Q = e\left(\frac{0.1723 \times T^4}{100{,}000{,}000}\right)$$

where

Q = total black body radiation, Btu per square foot per hour.
e = emissivity.
T = absolute temperature, degrees Fahrenheit.

between 142 Btu and the average amount of heat coming into the room is the amount which will be lost per square foot per hour by radiation from a body at 83 F. If the rate at which it is desired that heat be lost from the body by radiation and convection be assumed, the mean radiant emission from the walls required to give the desired result can be deter-

mined, as can also the required air temperature for the corresponding convective effect.

The determination of the amount of radiant heating surface needed in a room requires knowledge of the climate, the type of structure, the type of heating, and the surface temperature of the walls. This problem can be solved only on an empirical basis. After some experience, however, it is possible to estimate these variables with a considerable degree of accuracy for any climate or construction.

Assume that a mean radiant temperature of 65 F is desired. Table 1 shows that with all the walls at this temperature, and with an emissivity of 0.95, the gross heat emission is 124 Btu per square foot per hour. The total emission of radiation into the room from that surface would therefore be $A \times 124$, where A is the total inside area of the room. This is the *desired* emission.

If the whole area be divided into a number of different parts which are each at a uniform temperature—a_1, a_2, a_3, *etc.*,—and each is multiplied by the value of the heat emission corresponding to that temperature, and if all these products are added together, their sum will represent the total *actual* emission of radiation into the room at these temperatures without the aid of any hot surface.

The difference between the desired emission and the actual emission represents the additional heat which must be supplied by the hot surface. The temperature of the proposed hot surface must then be selected, and its emission per square foot at that temperature determined from Table 1. This emission is divided into the additional amount of heat needed, adjusted for the fact that the heating units will shield the walls behind them, and the quotient obtained will be the area of the required heating surface.

It is evident that this method of calculation is approximate, and depends for its accuracy on a correct estimate of the ultimate surface temperatures attained by the actual wall surfaces. The following example will illustrate the principles involved:

TABLE 2. SURFACE AREAS, TEMPERATURES AND EMISSIONS FOR A ROOM OF 5760 CU FT

	Area Sq Ft	Assumed Surface Temperature (Deg Fahr)	Heat Emission (Btu Per Sq Ft Per Hr)	Total Heat Emission from Area (Btu Per Hr)
External Wall	297	50	110.6	32,850
Glass	279	45	106.5	29,710
Inner Wall	480	55	115.1	55,250
Ceiling	480	55	115.1	55,250
Floor	480	55	115.1	55,250
Total	2016			228,310

Example 1. The surface areas, temperatures, and emissions for a room having a volume of 5760 cu ft are given in Table 2. The figures for temperatures are fairly representative of American practice with well built walls, and are based on an emissivity of 0.95 which approximates that of most paints and building materials.

The mean radiant temperature of the room is $\frac{228,310}{2016} = 113.2$ Btu per square foot

per hour which, as seen from Table 1, corresponds to an MRT of 53 F for an average emissivity of 0.95.

For an average individual having a body surface of 19.5 sq ft, under conditions of comfort with a body surface temperature of 83 F, the heat given off by radiation may be determined by means of Equation 1 as 217 Btu per hour, or 11.1 Btu per square foot per hour. This corresponds to an environmental emission of 142 − 11.1 = 130.9 Btu per square foot per hour, and according to Table 1, to an MRT of 72 F.

If this body be placed in the room described, it would lose heat at the rate of 19.5 (142 − 113.2) = 562 Btu per hour. This loss is 345 Btu per hour, or 17.7 Btu per square foot per hour, more than the rate of heat loss for comfort, which is only 19.5 (142 − 130.9) = 217 Btu per hour.

In order to determine the amount of radiating surface necessary to maintain the MRT at 72 F, assume the surface temperature of the hot plates to be installed to be 200 F, which is approximately the temperature they would have if heated by steam.

The 2016 sq ft total area of the surfaces of the room multiplied by 130.9, which is the emission in Btu per square foot per hour necessary to maintain a body surface temperature of 83 F, gives a total desired emission of 263,890 Btu per hour. It is necessary to supply enough radiant heating surface to increase the total actual mean radiant heat emission by the room from 228,310, as shown in Table 2, to the 263,890 Btu desired. The additional heat needed is the difference between these figures, or 35,580 Btu. Since, from Table 1, the emission per square foot at 200 F is 309 Btu, the required radiant heating surface needed is $\frac{35,580}{309}$ = 115 square feet. The effect of this surface suitably placed would be to raise immediately the mean radiant temperature to the required degree and to maintain it at that value as long as the surfaces remained at the values assumed.

It is necessary also to calculate how much heat will be given off by the same surfaces by convection, and thereby to determine whether this amount of convected heat will warm entering ventilating air to the temperature maintained. If it will not, additional convection surfaces must be introduced to make up the balance.

In the solution of this particular example, the radiation loss from the human body was selected as 217 Btu per hour, which is that taking place under optimum comfort conditions, with a body surface temperature of 83 F in a uniform environment at 72 F. The mean radiant temperature necessarily was 72 F. If the optimum BET of 72 deg Fahr is desired, an air temperature of 72 F also must be maintained. If it is desired to maintain a lower air temperature than this, a mean radiant temperature greater than 72 F must be selected and the radiation loss from the individual must be recalculated from Equation 1.

The calculation may be simplified by preparing tables showing, at the usual temperatures, the area of hot surface required to bring each square foot of actual wall surface at various temperatures up to a general standard of 60 F to 70 F. It would, therefore, be necessary only to multiply the respective areas by the appropriate factors, and to add the results, to obtain the required total.

MEASUREMENT OF RADIANT HEATING

Convection heating, having as its object the raising of the air temperature to a specified degree, must be measured by thermometric methods which indicate essentially the air temperature, and not the rate of heat loss from the human body. Radiant heating, having as its object the control of the rate of heat loss from the human body, can be measured only by methods which basically are calorimetric, that is, which measure

directly the rate of heat loss from an object maintained at the temperature of the body, irrespective of air temperature.

The apparatus for this purpose consists essentially of a hollow sphere, or cylinder, containing a fluid which can be maintained accurately at 83 F (the accepted mean surface temperature of the human body), with an accurate means of measuring the rate of heat supply required to maintain the temperature at that exact point. The latter measurement can be made with sufficient accuracy by electrical methods. Although a BET of 72 deg Fahr is desirable, the mean radiant and air temperatures may both vary, provided the heat loss by radiation and convection from a surface at 83 F is maintained at the rate of 15.4 Btu per square foot per hour, which corresponds to $\frac{15.4}{3.415} = 4.5$ watts per square foot of exposed surface.

This instrument, the *eupatheoscope*, can readily be adapted as a thermostat by electrical control to shut off or turn on heat when the critical temperature of 83 F in the vessel is increased or decreased. A modification of the instrument is called the *eupatheostat*.

Another instrument for maintaining comfort conditions is at present available only in a model adapted to British practice as it is designed for a temperature of 75 F. It consists of a blackened copper sphere of approximately 6 in. diameter in which is housed a cylindrical sump containing a volatile liquid. In operation, a small electric heating coil drawing about 5 watts creates in the sphere a vapor pressure which is constant as long as the heat losses from the sphere are standard. If the temperature of the air or the MRT becomes too high for comfort, a greater pressure is created, owing to a smaller loss of heat from the sphere. This increase of pressure acts on a diaphragm and shuts off the supply of heat to the room.

For testing work, the *globe thermometer* is a very useful instrument. It consists of an ordinary mercury thermometer, with its bulb placed in the center of a sphere about 6 in. to 9 in. in diameter, usually made of thin copper and painted black. The temperature thus recorded is termed the *radiation-convection temperature*.

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Chapter 38

ELECTRICAL HEATING

Resistors, Heating Elements, Electric Heaters, Unit Heaters, Central Fan Heating, Electric Steam Heating, Electric Hot Water Heating, Heat Pump, Control, Calculating Capacities, Power Problems, Electric Heating Data

WHILE it is improbable that electricity will ever replace fuels as the main source of heat, this type of heating has a logical and a rapidly growing place in the heating industry due to its advantages, such as flexibility, cleanliness, safety, convenience, and ease of control. Electric heating practice has many basic principles in common with fuel heating, but there are also important differences. The advantages of good building insulation are even more important in electric heating than for fuel heating, because the initial cost per Btu is usually higher.

All heat is a form of energy. Fuels hold stored chemical energy which is released into heat by combustion. Electrical power is a form of energy which can be released into heat by passing it through a resisting material. Both fuel and electric heating have two divisions: *first,* the conversion of energy into heat; *second,* the distribution and practical use of the heat after it is produced.

In converting the chemical energy of fuels into heat by combustion, there is necessarily a considerable variation in thermal efficiency. This is not true, however, when converting electric power into heat, because 100 per cent of the energy applied in the resistor is always transformed into heat. In electric heating practice the engineer need not be concerned about efficiencies of heat production, but rather about efficiencies of heat utilization.

DEFINITIONS

Definitions of terms used in fuel heating are given in Chapter 42. The following terms apply particularly to electric heating:

Electric Resistor: A material used to produce heat by passing an electric current through it.

Electric Heating Element: A unit assembly consisting of a resistor, insulated supports, and terminals for connecting the resistor to electric power.

Electric Heater: A complete assembly of heating elements with their enclosure, ready for installation in service.

RESISTORS

Solids, liquids, and gases may be used as resistors, but most commercial electric heating elements have solid resistors, such as metal alloys, and non-metallic compounds containing carbon. In some types of electric boilers, water forms the resistor and is heated by an alternating current of electricity passing through it.

HEATING ELEMENTS

Commercial electric heating elements are divided into open type elements, enclosed type elements, and cloth fabrics. *Open type elements* have resistors exposed to view. The resistors may be coils of wire or metal ribbon, supported by refractory insulation, or they may be non-metallic rods, mounted on insulators. Open type elements are used extensively for operation at high temperatures when radiant heat is desired. They are also frequently used at low temperatures for convection and fan circulation heating, especially in large installations.

Enclosed type elements have metallic resistors embedded in a refractory insulating material, and encased in a protective sheath of metal. Fins or extended surfaces may be used to add heat dissipating area. Enclosed elements are made in many forms, such as strips, rings, plates, and tubes. Strip elements are used for clamping to surfaces requiring heat by conduction, and in convection and fan circulation air heaters. Ring and plate elements are used in electric ranges, waffle irons, and in many small air heaters. Tubular elements may be immersed in liquids, cast into metal, and, when formed into coils, used in electric ranges and air heaters. *Cloth fabrics* woven from flexible resistor wires and asbestos thread, are used for many low temperature purposes.

ELECTRIC HEATERS

Electric heaters are classified according to the manner in which they deliver heat in practical use, that is, by conduction, by radiation, or by convection. The term *radiator* should not be used in electric heating, because of confusion between its established usage in fuel heating and the radiant principle of many electric heaters.

Among the uses of *conduction electric heaters*, which deliver most of their heat by actual contact with the object to be heated, are aviators' clothing, hot pads, foot warmers, soil heaters, ice melters, and pipe heaters. Conduction heaters are useful in conserving and localizing heat delivery at definite points. They are not suitable for general air heating.

Radiant electric heaters, which deliver most of their heat by radiation, have high temperature incandescent heating elements and reflectors to concentrate the heat rays in the desired directions. The immediate and pleasant sensation of warmth which is caused by radiant heat makes this type desirable for temporary use where the heat rays can fall directly upon the body. They are not satisfactory for general heating, as radiant heat rays do not warm the air through which they pass. They must first be absorbed by walls, furniture, or other solid objects which then give up the heat to the air. The location of radiant heaters is important. They should never face a window as some rays pass through glass and are lost. Figs. 1 and 2 show common types of portable and wall-mounted radiant heaters.

Convection electric heaters, designed to induce thermal air circulation, deliver heat largely by convection, and should be located and used in much the same manner as steam and hot water radiators or convectors. They should have heating elements of large area, with moderate surface temperature, enclosed to give proper stack effect to draw cold air from the floor line (Figs. 3 and 4). The flexibility possible with electric heating

elements should discourage the use of secondary mediums for heat transfer. Water and steam add nothing to the efficiency of an electric heater and entail expensive construction.

UNIT HEATERS

Fan unit electric heaters, having electric heating elements combined in the same enclosure with a fan or blower, are made in many styles and are excellent for general air heating. They should be located and used much as steam unit heaters. The warm air can be directed toward the floor, if desired, to give a positive circulation which will reduce stratification of

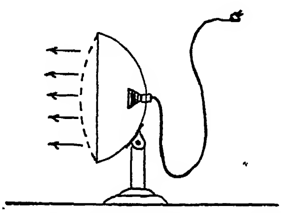


FIG. 1. PORTABLE RADIANT ELECTRIC HEATER

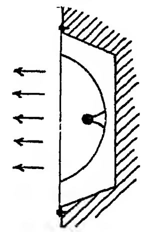


FIG. 2. RADIANT ELECTRIC HEATER RECESSED IN WALL

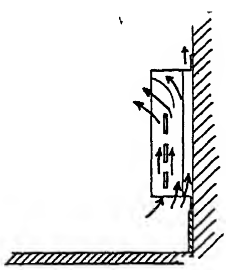


FIG. 3. CONVECTION ELECTRIC HEATER ON WALL SURFACE

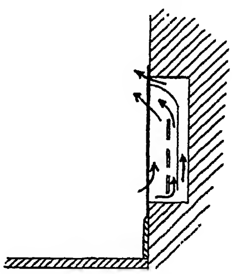


FIG. 4. CONVECTION ELECTRIC HEATER RECESSED IN WALL

air. Small units which are free from radio interference are used for homes; there are large units for industrial plants, substations, power houses, and pumping stations; portable units are useful for temporary work, such as drying out damp rooms, or for warming rooms during construction (Figs. 5, 6, 7 and 8).

CENTRAL FAN HEATING

Central fan electric heating systems have electric heating elements and fans or blowers to circulate the air through ducts, and in addition to the main heaters at the fan location, booster heaters may be located in branch ducts. Humidification or complete air conditioning can readily be included in the system, in much the same manner as with steam.

In coördinating the input of heat energy and the volume of air circulation, a basic difference between electric heating and steam heating enters into the problem. Steam is approximately a constant-temperature source of heat for any given pressure as a change in air volume flowing over steam coils does not greatly affect the temperature of the delivered

air. The amount of steam condensed (heat input) varies in proportion to the air volume, but the surface temperature of the steam coils remains about the same. Electric heat is quite different, being a constant source of energy. If the volume of air flow over electric heating elements is changed, and no change is made in the electrical power connections, there will be a corresponding change in the temperature of the air delivered because the electrical energy input remains constant and the surface temperature of the heating elements will vary as is necessary to force the air to accept all the heat. With electric heat the total heat is constant

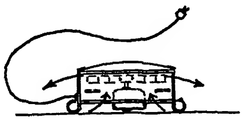


FIG. 5. SMALL PORTABLE FAN UNIT ELECTRIC HEATER

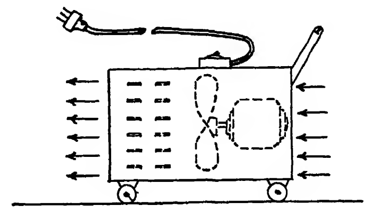


FIG. 6. LARGE INDUSTRIAL TYPE PORTABLE FAN UNIT ELECTRIC HEATER

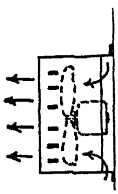


FIG. 7. SMALL FAN UNIT ELECTRIC HEATER

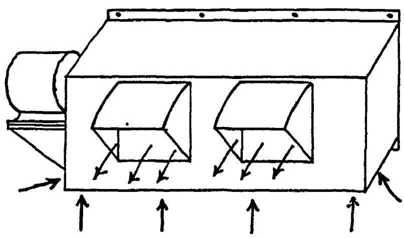


FIG. 8. LARGE INDUSTRIAL TYPE FAN UNIT ELECTRIC HEATER

unless some compensating action is performed by control. Automatic modulation to vary the electrical heat input and synchronize it properly with the air flow has been successfully applied to central fan systems.

ELECTRIC STEAM HEATING

Electric steam heating differs from fuel heating only in the use of *electric boilers* to generate steam. Small boilers usually have heating elements of the enclosed metal resistor type immersed in the water. Boilers of this construction may be used on either direct or alternating current since the heat is delivered to the water by contact with the hot surfaces. To lessen the likelihood that the heating elements will burn out, they are made removable for cleaning off deposits of scale which will restrict the heat flow. Boilers of this type are useful in industrial plants which require limited amounts of steam for local processes, and for sterilizers, jacketed vessels, and pressing machines which need a ready supply of steam.

Electric boilers are entirely automatic and are well adapted to intermittent operation. It frequently is economical to shut down the main plant boilers when the heating season ends, and to supply steam for summer needs with small electric boilers located close to the operation.

Large electric boilers are usually of the type employing water as the resistor. Only alternating current can be used, as direct current would cause electrolytic deterioration. Large boilers of this kind have electrodes immersed in the water where heat is generated directly. In Canada and Europe many successful installations have been made, but in the United States the cost of electric power, in comparison with fuels, does not favor their general use.

ELECTRIC HOT WATER HEATING

Electric hot water heating occupies much the same position as electric steam heating. It is useful in many moderate-sized installations, but large ones are seldom economical in this country. Electric boilers for supplying domestic needs for hot water are inexpensive, entirely automatic, and are insulated to prevent excessive heat losses. Similar boilers in large sizes are useful for industrial needs for hot water. When lower power costs can be secured, by confining the heating to certain fixed hours water may be heated and stored in well-insulated tanks for use when needed. In large industrial plants it is often possible to balance power loads by this means and to avoid running the fuel-fired steam boilers at night or over week ends. In Europe use has been made of this hot water storage principle for heating. Experiments have been made in this country for heating houses, but the cost of serving individual homes with the necessary heavy electric power loads has proved unprofitable at rates comparable to other forms of heating. The problems incident to installing large storage tanks in home basements, and the lack of flexibility under variable weather conditions, are also unfavorable factors.

OIL HEATING

Electric hot oil heating is useful in some industrial work as a substitute for superheated steam. Special oil can be electrically heated as high as 600 F and pumped at a pressure just sufficient to cause flow. When used in heating coils or jacketed vessels, this gives a safe, and convenient, automatic system for moderate-sized installations.

HEAT PUMP

The electric heat pump is not strictly an electric heater, as it does not directly convert electrical power into heat. It operates a compressor electrically which acts as a reversible refrigerating unit to extract heat from the outdoor air in winter and deliver it indoors for heating purposes, and, by a reversal, to extract heat from the indoor air in summer and discharge it outdoors. This system has been used in evenly-balanced climates where the heating requirements in winter are about the same as the cooling requirements in summer.

CONTROL

Because the efficiency of electric heat production is the same for large or small units, it is possible to reduce heat waste to a minimum by applying local heating, locally controlled. Wherever radiant heaters are used, thermostats are not an effective means of control and manual operation or control by eupatheoscope is necessary. For all convection and fan circulation heaters thermostatic control is useful. For small heaters having ratings up to about 1500 watts, there are direct-acting thermostats which are satisfactory, but for larger heaters it is advisable to use relays

or contactors, which should break all of the power lines. All heaters having fan circulation should have the heat circuit interlocked with the motor circuit so that the fan will be running when the heat is on. A thermal fuse or trip should be located in the heat chamber to throw off the heat in case any interruption of air flow should occur; otherwise undue temperature rise would result. In all large heaters the heating elements should be arranged in groups and control provided to vary the heat input to correspond approximately to the heat demand. If this is not done, and all the heat is kept available, the thermostat will continue throwing it on and off at short intervals. Except for central fan systems, the heat stages can be operated by manual switches, but automatic modulation of the heat load is usually preferred.

CALCULATING CAPACITIES

The methods of calculating heat losses outlined in Chapters 6, 7, and 8 may be used for electric heating exactly as for fuel heating. The total heat requirements in Btu per hour may then be converted into the electrical rating of an equivalent heating system by using the equation:

$$\frac{\text{Total Btu per hour}}{3415} = \text{kw rating of required electric heating} \qquad (1)$$

POWER PROBLEMS

The first point to determine is the cost of the power which is available for electric heating. Unlike fuels, there is no uniform cost for electric power because of the unequal cost of distribution to large and small users. The fact that electricity cannot be economically stored, but must be used as fast as it is generated, makes it impossible to operate power plants at uniform loads; hence, even the time of use may affect the cost of power. As distribution is a big item in power costs, the best places to use power for heating are large industrial plants which have heavy service lines.

Homes are almost universally supplied with lighting current of 115 volts, which cannot be used economically for any but the smallest heaters. Usually the service lines will not permit more than plug-in devices. The underwriters permit heaters of 1250 watts to be used from approved base board receptacles. Where homes have 230 volt service for cooking and water heating, and rates are favorable, larger heaters can be installed. For industrial purposes, heaters should be designed to use polyphase power, which is usually supplied at 230, 460 or 575 volts. All polyphase heaters should be balanced between phases.

ELECTRIC HEATING DATA

Electric heater capacity is rated in kilowatts (kw). Electric power is measured in kilowatt-hours (kwh). Cost of operation = kw rating × hours used × cost per kwh.

One boiler horsepower (bhp) = 33,471.9 Btu per hour
One kilowatt-hour (kwh) = 3,415 Btu per hour

$$\text{One boiler horsepower} = \frac{33{,}471.9}{3{,}415} = 9.80 \text{ kwh}$$

One boiler horsepower will evaporate 34.5 lb water per hour *from and at* 212 F.

$$\text{One kilowatt-hour} = \frac{34.5}{9.80} = 3.52 \text{ lb of water per hour at 212 F}$$

Additional conversion factors are given in Chapter 42.

Chapter 39

WATER SUPPLY PIPING

Maximum Possible Flow, Maximum Probable Flow, Average Probable Flow, Factor of Usage, Kind of Pipe Used, Sizing of Risers, Sizing of Mains, Sizing of Systems, Hot Water Supply, Hot Water Storage

IN the design and layout of domestic water supply systems, the engineer is confronted with the necessity of combining the somewhat empirical rules and formulae in use with the more or less exact hydraulic principles involved. Unlike heating and ventilating layouts, there are practically no definite data for estimating the quantity of water likely to be consumed or the probable rate of water flow at any particular moment.

Metered results in one building often show two or three times the metered amount in another building of the same size and with the same type of tenants. In hotels, one riser will often have an almost constant flow that may never be reached by another at peak load. In office buildings, the women's toilets show a far greater daily consumption than those of the men, yet at no time will they approach the hourly consumption of the men's toilet during the first hour of the day. This condition has led to a multiplicity of rules of practice which vary as much as the data used. All must of necessity be based on an assumed rate of consumption and on an assumed probability of simultaneous use, and while the formulae employed may have been derived on sound technical bases the assumptions are often in error.

To arrive at a safe standard, the approximate rate of flow of each fixture to be supplied must be known and the probable number of fixtures in use at any one time must be assumed. Obviously, the maximum number of fixtures assumed to be in use must be taken at the peak of demand and the lines must be made adequate to supply such a peak regardless of the riser or branch on which the demand may occur. This means that all water piping under the usual conditions will be over-sized.

In tall buildings it is customary to divide the water supply systems, both hot and cold, into sections of 10 to 20 stories. Such *zoning* or *sectionalizing* is for the purpose of avoiding excessive pressures on the fixtures in the lower stories of each system. This limits the consideration of water pipe sizes to horizontal mains and to risers not exceeding 20 stories in height or about 200 ft[1].

[1]It is impractical to attempt to size piping so as to produce the proper pressure on fixtures at different levels by employing friction, owing to the fact that this friction will be built up to the amount desired only in times of maximum demand and at all other times the friction will be only a fraction of the maximum friction so that the fixtures by this method are subjected to a varying pressure on the water supply line. A much more practical method is to throttle the flow at the fixture, or to use flow regulators, so that the quantity of water delivered will approximate the fixture demands and so that this is accomplished without splashing or noise.

For the purpose of this chapter the following terms will be used and should be clearly distinguished from one another:

Maximum Possible Flow: The flow which would occur if the outlets on all fixtures were opened simultaneously. This condition is seldom, if ever, obtained in actual practice except in cases of gang showers controlled from one common valve, and similar conditions.

Maximum Probable Flow: The maximum flow which any pipe is likely to carry under the peak conditions. This is the most important amount to be considered in pipe sizing.

Average Probable Flow: The flow likely to be required through the line under normal conditions.

It is evident that any pipe adequate to take care of the *maximum*

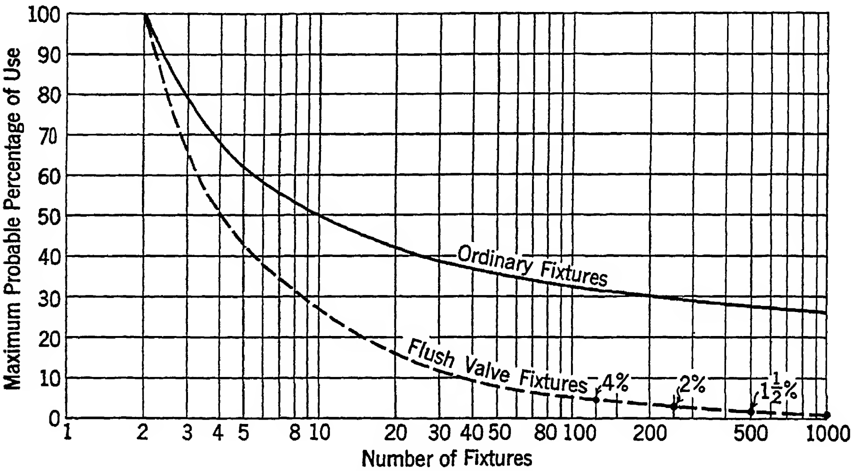


FIG. 1. CHART SHOWING RELATION BETWEEN NUMBER OF FIXTURES AND MAXIMUM PROBABLE PERCENTAGE OF USE

probable flow will also be more than able to take care of the *average probable flow*, and hence the latter has no bearing on the pipe size.

MAXIMUM PROBABLE FLOW

There are two factors to be considered in calculating the maximum probable flow, namely, (1) the quantity of water that will flow from the outlets when they are open, and (2) the number of outlets likely to be open *at the same time*. Table 1 shows the maximum approximate rate of flow from each fixture when it is in use, and will serve as a guide in estimating maximum probable flow demands although there is considerable variation in different fixtures and valves. Probably the flow under normal water pressures, or with the pressure properly throttled, will not differ greatly from the values stated. With the aid of this table it is possible to calculate the maximum possible flow with all outlets open in both the hot and cold water lines.

Factor of Usage

To obtain the maximum probable flow it is necessary to multiply the maximum possible flow by a factor of usage, and this factor varies with the installation and the number of fixtures in the installation. It is evident that with two fixtures it is quite possible that both will at some time be in operation simultaneously. With 200 fixtures, it is unlikely the entire 200 would ever operate at the same time. Consequently, the factor of usage reduces as the number of fixtures becomes greater, all other things being equal. On the other hand it is probable that outside of flush valve

TABLE 1. APPROXIMATE FLOW FROM FIXTURES UNDER NORMAL WATER PRESSURES

FIXTURES	COLD WATER (GALLONS PER MINUTE)	HOT WATER (GALLONS PER MINUTE)
Water-closets, flush valve	50[a]	0
Water-closets, flush tank	18	0
Urinals, flush valve	40[a]	0
Urinals, flush tank	18	0
Urinals, automatic tank	1	0
Urinals, perforated pipe per foot	10	0
Lavatories	3	3
Showers, 5 to 6½ in. heads	3	3
Showers, tubular	6	6
Needle bath	30	30
Shampoo spray	1	1
Liver spray	2	2
Manicure table	1½	1½
Baths, tub	5	5
Kitchen sink	4	4
Pantry sink, ordinary	2	2
Pantry sink, large bibb	6	6
Slop sinks	4	4
Wash trays	3	3

[a]Actual tests on water-closet flush valves indicate 40 gpm as the maximum rate of flow with 30 lb pressure at the valve; this would increase to 60 gpm (about 50 per cent) at 90 lb pressure. The 50 gpm has been taken as an average flow; possibly, with very low pressures just sufficient to operate the flush valve, 30 gpm could be allowed with safety. Urinal flush valves would vary proportionately in the same manner.

fixtures, the factor of usage would never be less than about 25 per cent no matter how many fixtures were installed, provided no fixtures in excess of those required for the actual occupancy were included.

This factor, beginning at 100 per cent for two ordinary fixtures, decreases rapidly until 5 fixtures is reached and then becomes almost constant, as shown in the upper curve, Fig. 1. This applies to a normal building and not to institutions where the inmates may all be required, for instance, to bathe on certain days of the week and at certain hours of those days. In such special cases a new factor of usage must be developed based on the maximum probable usage under the conditions involved. For flush valve fixtures the quantity of water is greater, but owing to the short duration of the flush, the simultaneous usage drops more rapidly so as to reach 1 per cent for 1000 fixtures as shown, on lower curve, Fig. 1[2].

[2]This can be proved by assuming, for example, 1000 water-closets which would not be used more than six times per hour (or once every 10 minutes) and which require from 5 to 7 gal per flush or an average of about 6 gal. If these closets were all being used at their utmost capacity, the water demand would be 600 gpm. But average use would be about one-third of this and peak conditions would be in the neighborhood of twice the average, or about 400 gpm as the maximum that would ever develop. Assuming 50 gpm as the maximum rate of flow per closet and 1 per cent of the total closets in operation, the rate would be 50 gpm × 1 per cent of 1000 or 500 gpm. This is 100 gpm higher than obtained by the first method indicating an additional factor of safety over the first method.

Example 1. Assume that in a normal building, such as a residential hotel or an apartment house, there are 50 flush valve water-closets, 50 lavatories, 50 sinks and 50 baths, and that it is desired to determine the maximum probable flow in a line supplying all of these fixtures with both hot and cold water. Fig. 1 shows a maximum probable use for 50 water closets of about 8 per cent and for 150 ordinary fixtures, of about 31 per cent. Therefore:

Cold Water

50 W. C. x 50 gpm at 8 per cent		200 gpm
50 Lavs. x 3 gpm	150 gpm	
50 Sinks x 4 gpm	200 gpm	
50 Baths x 5 gpm	250 gpm	
150 Fixtures	600 gpm at 31 per cent	186 gpm
Total maximum probable flow of cold water		386 gpm

Hot Water

50 W. C.		None
50 Lavs. x 3 gpm	150 gpm	
50 Sinks x 4 gpm	200 gpm	
50 Baths x 5 gpm	250 gpm	
150 Fixtures	600 gpm at 31 per cent	186 gpm
Total for main supplying cold and hot water		572 gpm

It should be noted that this is a *rate of flow* or an *instantaneous demand.*

KIND OF PIPE USED

Before entering into the actual sizing of pipe, it is necessary to consider the kind of pipe to be used and to make suitable allowance for corrosion and fouling during the lifetime of the system. For example, if brass, copper or alloy pipe is contemplated, it is probable that the quantities indicated in Example 1 are ample; if galvanized pipe is to be used, then it is quite likely that after a period of say 15 years the area may be decreased as much as 25 per cent and the quantitities of water assumed should be increased by 35 per cent to allow for this reduction of area; if the water contains lime it is possible that 50 per cent of the area may be lost and in such cases the flow should be doubled and no branch pipe connected to fixtures should be less than ¾ in. In all of the following calculations, the assumption is made that the water is fairly good and that a corrosion resistant type of pipe is to be used.

SIZING A DOWN-FEED RISER

Down-feed systems are commonly used for tall buildings. In sizing a riser arranged for down-feed, the gravity head permits a pressure drop that is almost prohibitive in an up-feed riser. There is a gain in riser head of 0.43×100 or 43 lb per 100 ft of run and hence it is quite permissible to size such a riser on the basis of a pressure drop of 30 lb per 100 ft of run, as the difference between the 43 lb generated and the 30-lb drop under maximum probable demand is ample to take care of the friction caused by

the fittings. This method applied to the typical riser shown in Fig. 2 gives the schedule of sizes indicated in Table 2 for any flow from 5 to 250 gal.

TABLE 2. SCHEDULE OF SIZES FOR DOWN-FEED RISER. (SEE FIG. 2)

Portion of Riser	Maximum Probable Flow, Gallons per Minute 5	10	15	20	25	30	40	50	60	70	80	90	100	125	150	200	250
T	¾	1	1¼	1¼	1½	1½	2	2	2½	2½	2½	2½	2½	3	3	3½	3½
S	¾	¾	1	1	1	1¼	1¼	1½	1½	1½	2	2	2	2	2½	2½	2½
R	¾	¾	¾	1	1	1	1¼	1½	1½	1½	2	2	2	2	2	2½	2½
Q	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
P	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
O	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
N	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
M	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
L	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
K	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
J	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
I	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
H	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
G	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
F	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
E	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
D	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
C	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
B	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½
A	¾	¾	¾	1	1	1	1¼	1½	1½	1½	1½	2	2	2	2	2½	2½

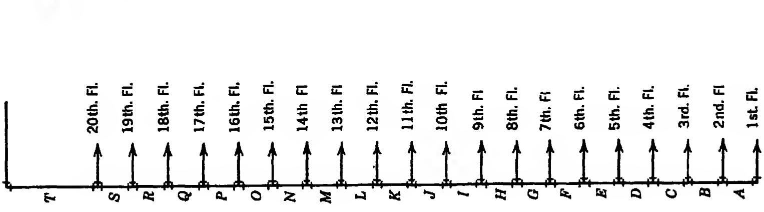


FIG. 2. TYPICAL RISER FOR 20-STORY BUILDING

SIZING AN UP-FEED RISER

When the riser is an up-feed, the opposite condition occurs, that is, there is a drop in pressure as the top of the riser is approached, due to the natural reduction in the gravity pressure, and to this must be added the

pipe friction plus that introduced by the pipe fittings, all of which produce an excessive drop when compared to the conditions existing with a down-feed riser.

To size an up-feed riser the minimum pressure of the street main, or other source of supply, should be ascertained and from this should be subtracted the pressure to be maintained at the highest fixture, namely, 15 lb per square inch, plus the height in feet above the source of water pressure, multiplied by 0.43 to change from feet of head to pounds of pressure. The total length of run from the source of pressure to the farthest and highest fixture should be ascertained, and this should be changed to equivalent length of run to allow for the loss occasioned by the pipe fittings. Table 3 gives the additional lengths necessary to allow for the various fittings and valves. The drop allowable in pressure per 100 ft of run may then be obtained by multiplying the surplus pressure (over that required for the gravity head and to supply 15 lb at the fixture) by 100 and by dividing this by the equivalent length of run to the farthest or highest fixture.

Example 2. Assume a street pressure of 60 lb, the height of the highest fixture 50 ft, and the length of the longest run 200 ft. Without knowing the additional length of pipe to be added for the fittings it will be assumed that this is about 100 ft. The surplus pressure which will be available for pressure drop will then be

$$60 \text{ lb} - (15 \text{ lb} + 50 \text{ ft} \times 0.43 \text{ lb}) =$$
$$60 \text{ lb} - (15 \text{ lb} + 21.5 \text{ lb}) = 23.5 \text{ lb}$$

To change this into drop per 100 ft:

$$\frac{23.5 \text{ lb} \times 100}{200 \text{ ft} + 100 \text{ ft}} = 7.8 \text{ lb per 100 ft.}$$

The pipe may then be sized from the maximum probable flow by selecting a size that does not give a drop in excess of 7.8 lb per 100 ft.

It will be seen from Example 2 that it is impossible to size up-feed risers without determining the drop allowable in both the horizontal feed mains and the toilet room branches. Having once ascertained this allowable drop, it is simply a matter of applying it throughout the system.

TABLE 3. APPROXIMATE ALLOWANCES FOR FITTINGS AND VALVES IN FEET OF STRAIGHT PIPE

Size of Pipe (Inches)	Type of Fitting or Valve					
	90-Deg Elbow	45-Deg Elbow	Return Bend	Gate Valve	Globe Valve	Angle Valve
½	4	3	8	2	48	8
¾	5	3	10	3	60	10
1	5	3	10	3	60	10
1¼	6	4	12	3	72	12
1½	7	5	14	4	84	14
2	7	5	14	4	84	14
2½	10	7	20	5	120	20
3	12	8	24	6	144	24
4	18	13	36	9	216	36
5	25	18	50	13	300	50
6	30	21	60	15	360	60

HORIZONTAL SUPPLY MAINS

The horizontal mains supplying the risers at the top of a down-feed system must be liberally sized unless the house tank is set at a much higher elevation than usual. To provide a gravity head on the highest fixtures of 15 lb per square inch it is necessary for the water line in the house tank to be nearly 40 ft higher, and with the line loss considered this becomes about 45 ft. Such heights are not often practical and as a result the pressure on the highest fixtures either is reduced to 7 lb (which is sufficient to operate a flush valve), flush tank water-closets are substituted, or a separate cold and hot water supply is installed with a small pneumatic tank to give the increase in pressure necessary. The chief objection to the use of a pneumatic tank is that a separate hot water heater is required and this heater must be located either sufficiently below the highest fixtures to obtain a gravity circulation, or it must be provided with a circulating pump in order to force the hot water to the top floor level.

The most common solution is to place the house tank as high as the structural and architectural conditions will permit and then to use liberally-sized lines between the house tank and the upper fixtures, say for the two top stories, below which the riser sizes may be reduced to those indicated in Fig. 2 and Table 2. Where the house tank is only one story above the top fixtures, flush tank water-closets must be used and the drop in the entire run from the house tank down to the farthest fixture should not exceed 1 lb; the less, the better. This means that if the total equivalent run to the farthest top fixtures supplied is 300 ft, the drop per 100 ft should not exceed $\frac{1\text{ lb} \times 100}{300}$ or 0.33 lb per 100 ft. The friction curves shown in Fig. 3 may be used for quickly determining the proper size of pipe to give any desired drop in pounds per 100 ft of equivalent run.

SIZING AN OVERHEAD DISTRIBUTION MAIN

Example 3. Suppose an installation has a house tank in which the water line is 20 ft above the level of the top fixtures to be supplied and that the length of run to the farthest fixtures on this level is 400 ft with the pipe fittings adding another 200 ft, making an equivalent length of 600 ft. What would be the size of main coming out of the tank where a maximum flow rate of 400 gpm may be expected, of the horizontal main where a maximum flow rate of 200 gpm may be expected and of the riser down to the fixture level where the maximum flow rate is approximately 100 gpm?

Here the level of the water in the house tank is 20 ft above the faucet of the highest fixture and the gravity pressure will be

$$0.43\text{ lb} \times 20\text{ ft} = 8.6\text{ lb}$$

and, if a total pressure drop of 1 lb is assumed, the pressure on the farthest fixture under times of peak load will be

$$8.6\text{ lb} - 1\text{ lb} = 7.6\text{ lb}$$

while the drop per 100 ft of equivalent run will have to be

$$\frac{1\text{ lb} \times 100}{600} = 0.1667\text{ lb}.$$

Referring to Fig. 3 it will be noted that where the flow through the main is 400 gpm, an 8 in. pipe would be required; that where the flow is reduced to 200 gpm, a 6-in. pipe

would be sufficient, and that where the flow is 100 gpm in the riser branch and riser, a 5-in. size would be correct. Of course these are somewhat excessive flows and the head from the tank is small so that large sizes are to be expected. It would be necessary to carry a 5-in. riser down to the branch to the top floor, then reduce to 4 in. for the branch to the floor below the top, and below this the sizes in Table 2 could be followed. In such a case, flush tank closets should doubtless be substituted.

Had the tank been set 10 ft higher, the head available to be used up in friction, but

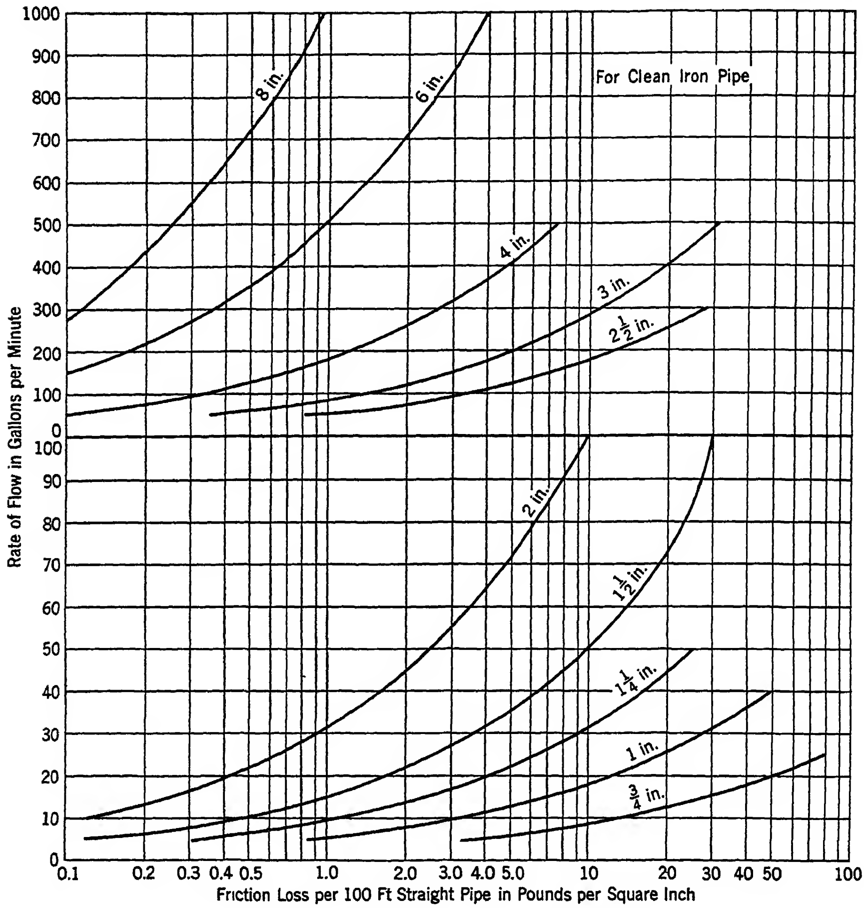


Fig. 3. Chart Giving Friction Losses for Various Rates of Flow of Water

still giving the same pressure at the top fixtures, would have been 0.43 lb × 10 ft or 4.3 lb greater and this, with the 1 lb drop used previously, would give a total allowable drop of

$$1 \text{ lb} + 4.3 \text{ lb or } 5.3 \text{ lb}$$

which, divided by the 600 ft equivalent run gives a drop per 100 ft of

$$\frac{5.3 \times 100}{600} \text{ or } 0.9 \text{ lb}$$

and, with this drop, the sizes according to the chart (Fig. 3) are 6 in., 4 in., and 4 in.,

respectively, while if the run is reduced to 200 ft instead of 600 ft, the allowable drop will be

$$\frac{5.3 \text{ lb} \times 100}{200} \text{ or } 2.7 \text{ lb per } 100 \text{ ft.}$$

This gives 5 in., 4 in., and 3 in., respectively, for the flows of 400, 200, and 100 gpm.

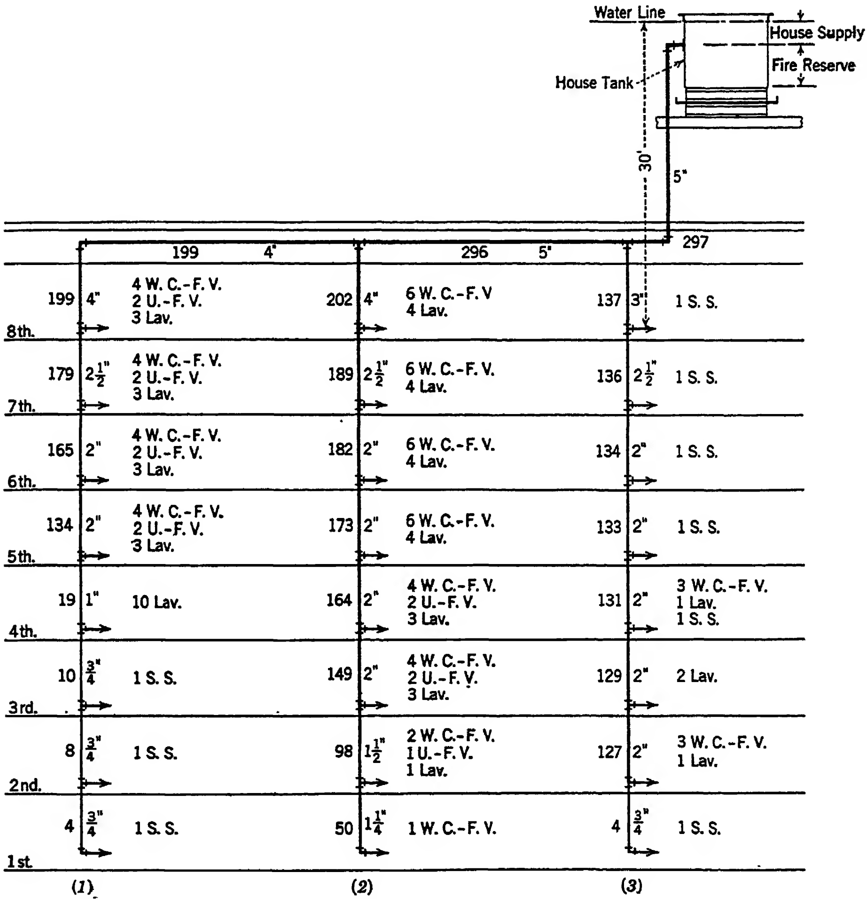


FIG. 4. TYPICAL LAYOUT FOR DOWN-FEED SYSTEM

From Example 3 it is evident that, while the down-feed system possesses certain economies in size for the riser portion, it is quite likely to involve large distribution main sizes especially when the tank is not elevated to a considerable degree.

SIZING A PIPING SYSTEM

Example 4. Fig. 4 shows a typical layout with three risers extending eight stories and with the fixtures noted on each floor. First this will be solved for a down-feed arrangement assuming that the level of the water in the house tank is 30 ft above the fixtures on

the top floor, that the length of run from the tank to the farthest fixture is 200 ft, equivalent length of fittings 100 ft, and the pressure required at the fixture is 7 lb.

The 30-ft head is equal to a static pressure of 0.43 × 30 or 12.9 lb per square inch and to maintain a pressure of 7 lb at the highest fixtures the drop allowable in pressure is 12.9 − 7.0 lb or 5.9 lb. As the total equivalent run is 300 ft, this is a drop per 100 ft of 1.97 lb, or practically 2 lb. Therefore, all risers and mains from the top floor back to the

Table 4. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

Riser No. 1

Floor of Bldg.	Fixtures on Floor	GPM per Fixt.	Max. Fixt. GPM	Probable Use (per cent)	Probable Fixt. GPM	Probable Riser GPM	Allowable Drop Lb per 100 ft	Pipe Size In.
1st	1 S. S.	4	4	100	4	4	30	¾
2nd	1 S. S.	4	4	100	4			
						8	30	¾
3rd	1 S. S.	4	4					
	3		12	80	10	10	30	¾
4th	10 Lav.	3	30					
	13		42	45	19	19	30	1
5th	4 W. C.	50	200					
	2 U.	40	80					
	6		280	40	112			
	3 Lav.	3	9					
	16		51	45	22	134	30	2
6th	4 W. C.	50	200					
	2 U.	40	80					
	12		560	25	140			
	3 Lav.	3	9					
	19		60	42	25	165	30	2
7th	4 W. C.	50	200					
	2 U.	40	80					
	18		840	18	151			
	3 Lav.	3	9					
	22		69	40	28	179	30	2½
8th	4 W. C.	50	200					
	2 U.	40	80					
	24		1120	15	168			
	3 Lav.	3	9					
	25		78	40	31	199	2	4

tank must be sized on the basis of a drop of 2 lb per 100 ft. Tables 4, 5, 6 and 7 show the schedule for Risers Nos. 1, 2 and 3 with the maximum possible flow taken from Table 1, the percentage of use at the peak taken from Fig. 1 and the maximum probable flow at the peak worked out for each portion of the riser, the riser sizes being taken from Table 2 as far as possible and from Fig. 3 where the amounts exceed the values given in this table; a drop of 30 lb per 100 ft is used except on the riser from the top floor back to the tank where 2 lb per 100 ft is the allowable limit.

The reduction in pipe size which would occur if flush tank water-closets were used on the top floor and only 3 lb pressure used on the fixtures is given in Tables 8 and 9.

This illustrates why flush tank closets so frequently are substituted on the uppermost floor when a house tank is the source of water pressure.

If it is now assumed that Riser No. 1 is to be fed from the bottom and the minimum street pressure is 75 lb with the top fixture of the riser 80 ft above the main, the problem

TABLE 5. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

Riser No. 2

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXT.	MAX. FIXT. GPM	PROBABLE USE (PER CENT)	PROBABLE FIXT. GPM	PROBABLE RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
1st	1 W. C.	50	50	100	50	50	30	1¼
2nd	2 W. C.	50	100					
	1 U.	40	40					
	4		190	50	95			
	1 Lav.	3	3	100	3	98	30	1½
3rd	4 W. C.	50	200					
	2 U.	40	80					
	10		470	30	141			
	3 Lav.	3	9					
	4		12	70	8	149	30	2
4th	4 W. C.	50	200					
	2 U.	40	80					
	16		750	20	150			
	3 Lav.	3	9					
	7		21	70	14	164	30	2
5th	6 W. C.	50	300					
	22		1050	15	157			
	4 Lav.	3	12					
	11		33	48	16	173	30	2
6th	6 W. C.	50	300					
	28		1350	12	162			
	4 Lav.	3	12					
	15		45	45	20	182	30	2
7th	6 W. C.	50	300					
	34		1650	10	165			
	4 Lav.	3	12					
	19		57	42	24	189	30	2½
8th	6 W. C.	50	300					
	40		1950	9	175			
	4 Lav.	3	12					
	23		69	40	27	202	2	4

would be solved by determining the maximum rate of flow in each portion of the riser as shown in Table 10 and then finding the allowable drop which can be used per 100 ft. The 80 ft of riser height will use up

$$0.43 \text{ lb} \times 80 \text{ or } 34.4 \text{ lb}$$

and the pressure at the top of the required 15 lb will make the total reduction 49.4 lb, leaving a balance of 25.6 lb which may be used up in friction. If the distance from the

Table 6. Typical Calculation of Pipe Sizes on Down-Feed Riser with Flush Valve Water-Closets and Urinals

Riser No. 3

Floor of Bldg.	Fixtures on Floor	GPM per Fixt.	Max. Fixt. GPM	Probable Use (per cent)	Probable Fixt. GPM	Probable Riser GPM	Allowable Drop Lb per 100 ft	Pipe Size In.
1st	1 S. S.	4	4	100	4	4	30	¾
2nd	3 W. C.	50	150	80	120			
	1 Lav.	3	3					
	2		7	100	7	127	30	2
3rd	0 W. C.	0	000					
	3		150	80	120			
	2 Lav.	3	6					
	4		13	70	9	129	30	2
4th	3 W. C.	50	150					
	6		300	40	120			
	1 Lav.	3	3					
	1 S. S.	4	4					
	6		20	55	11	131	30	2
5th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	7		24	53	13	133	30	2
6th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	8		28	51	14	134	30	2
7th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	9		32	50	16	136	30	2½
8th	0 W. C.	0	000					
	6		300	40	120			
	1 S. S.	4	4					
	10		36	48	17	137	2	3

street main to the bottom of the riser, which will be assumed to be the farthest one on the horizontal line, is 100 ft, and if the fittings are sufficient to add another 100 ft, as well as the 80 ft of vertical distance up the riser, the total equivalent run will be 280 ft, which will be taken as even 300 ft. Then the allowable drop per 100 ft will be

$$\frac{25.6 \text{ lb} \times 100}{300} \text{ or } 8.5 \text{ lb}$$

and the sizes shown in Fig. 5 are based on this amount of drop. Of course the other

risers will have the same maximum flows at the bottom as they formerly had at the top, namely 202 and 137 gal, respectively, for Risers Nos. 2 and 3. Combining these maximum flows in the same manner as pursued in the down-feed system it is seen that the maximum flow between Riser No. 2 and Riser No. 3 is 296 gpm, and between Riser No. 3 and the street main, 297 gpm which at a drop of 8.5 lb gives the main sizes indicated. It will be noted that in determining the maximum flow in an up-feed riser

TABLE 7. SIZE OF DISTRIBUTION MAIN FOR DOWN-FEED SYSTEMS. (SEE FIG. 4)

Riser No.	Fixtures	GPM per Fixture	Maximum Fixture GPM	Probable Use (Per Cent)	Probable GPM	Allowable Drop (Lb per 100 Ft)	Size of Main (Inches)
	16 W. C.	50	800				
	8 U.	40	320				
	24		1120	15	168		
1	22 Lav.	3	66				
	3 S. S.	4	12				
	25		78	40	31		
					199	2	4
	35 W. C.	50	1750				
	5 U.	40	200				
	64		3070	8	245		
2	23 Lav.	3	69				
	48		147	35	51		
					296	2	5
	6 W. C.	50	300				
	70		3370	7	236		
	4 Lav.	3	12				
3	6 S. S.	4	24				
	58		183	33	61		
					297	2	5

it is necessary to begin at the top floor and work down instead of beginning at the bottom floor and working up as was done in the down-feed sizing.

SIZING UP-FEED AND DOWN-FEED HOT WATER SYSTEMS

Hot water supply systems, when of the circulating type, have a few differences to be considered although the same general principles of sizing apply to these lines as to the cold water lines. Owing to the fact that there are no flush valves on the hot water piping and also on account of the many plumbing fixtures having no hot water connections, the sizes of the hot water piping in general will be considerably less than the cold water piping in the same building. On the other hand it is almost invariably required that a gravity circulation be kept up in such hot water lines and this often has a considerable influence on the size. There are three methods of arranging circulation lines, as follow:

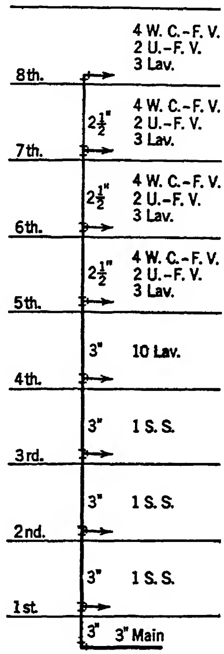
1. By using the plain up-feed with a return carried back from the top of the riser and paralleling it.
2. By carrying a supply riser up in one location thus supplying fixtures on up-feed, then crossing over at the top and coming down past another collection of fixtures and supplying these by a down-feed.
3. By carrying all of the water to the top of the building and dropping risers wherever needed, feeding all hot water on a down-feed system.

TABLE 8. TYPICAL CALCULATION OF PIPE SIZES ON DOWN-FEED RISERS WITH FLUSH TANK WATER-CLOSETS AND URINALS ON TOP FLOOR ONLY

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXT.	MAX. FIXT. GPM	PROBABLE USE (PER CENT)	PROBABLE USE GPM	PROBABLE RISER GPM	ALLOWABLE DROP LB PER 100 FT	PIPE SIZE IN.
				Riser No. 1				
7th and below	12 W. C.	50	600					
	6 U.	40	240					
	18		840	18	151			
	19 Lav.	3	57					
	3 S. S.	4	12					
	22		69	40	28	179	30	2½
8th	0 W. C.	00	000					
	0 U.	00	000					
	18		840	18	151			
	4 W. C.	18	72					
	2 U.	18	36					
	3 Lav.	3	9					
	31		186	37	69	220	3.3	4
				Riser No. 2				
7th and below	29 W. C.	50	1450					
	5 U.	40	200					
	34		1650	10	165			
	19 Lav.	3	57	42	24	189	30	2½
8 th	0 W. C.	00	000					
	0 U.	00	000					
	34		1650	10	165			
	6 W. C.	18	108					
	4 Lav.	3	12					
	29		177	38	67	232	3.3	4
				Riser No. 3				
7th and below	6 W. C.	50	300	40	120			
	5 S. S.	4	20					
	4 Lav.	3	12					
	9		32	50	16	136	30	2½
8th	0 W. C.	00	000					
	6		300	40	120			
	1 S. S.	4	4					
	10		36	48	17	137	3.3	3

The last method is usually the most satisfactory. (See Fig. 6).

In the first instance the up-feed riser may be sized for the same pressure drop as used for the cold water riser and, from the top of the riser *just below the top fixture connection*, a return circulation line may be carried back to the main return line in the basement and connected through a check valve set on a 45-deg angle and a gate valve; these return circulation lines should never be less than ¾ in., and on the farther half of the risers, not less than 1 in. to favor circulation in the far end. Typical top and bottom connections for such risers are shown in Fig. 7.



(1)

FIG. 5. UP-FEED SYSTEM

For the second arrangement of hot water risers, circulation lines are run back from the last fixture supplied to the main return circulation line in the same manner as just described, using ¾ in. for the near risers and 1 in. for the far risers. The sizing is much more difficult, as it is necessary to start at the bottom floor of the return riser and work back to the top of this riser and then carry the maximum flow across onto the top of the corresponding supply riser and work down on this riser from the top floor to the bottom. Naturally this gives a much greater flow in the supply riser and aids circulation by reducing pipe friction. The allowable loss per 100 ft in such lines must be made about half that used for the cold

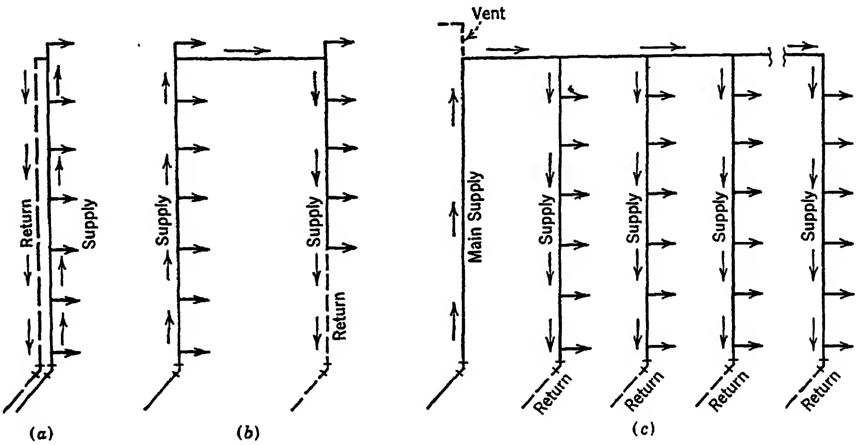


FIG. 6. METHODS OF ARRANGING HOT WATER CIRCULATION LINES

water risers which do not have the combined up- and down-travel which the hot water must make.

In the third and most common arrangement all of the water is carried from the tank or heater directly to the top of the building and is there distributed to the risers which are down-feed and may be sized in the regular down-feed manner if the total equivalent run either from the street main or house tank is taken into consideration. The return circulation lines from the botton of each riser should be arranged in the manner already outlined and any riser not going to the basement to

TABLE 9. SUMMARY OF RISER SIZES TO GIVE MAIN SIZES

Riser No.	Fixtures	GPM per Fixture	Riser GPM	Probable Use (Per Cent)	Probable GPM	Allowable Drop (Lb per 100 Ft)	Size of Main (Inches)
1	12 W. C.	50	600				
	6 U.	40	240				
	18		840	18	151		
	31 Fixt.		186	37	69		
					220	3.3	4
2	29 W. C.	50	1450				
	5 U.	40	200				
	52		2490	8	199		
	29 Fixt.		177				
	60		363	33	120		
					319	3.3	4
3	6 W. C.	50	300				
	58		2790	7	195		
	10 Fixt.		36				
	70		399	33	131		
					326	3.3	4

TABLE 10. TYPICAL CALCULATION OF PIPE SIZES ON UP-FEED RISER WITH FLUSH VALVE WATER-CLOSETS AND URINALS

FLOOR OF BLDG.	FIXTURES ON FLOOR	GPM PER FIXT.	MAX. FIXT. GPM	PROBABLE USE (PER CENT)	PROBABLE FIXT. GPM	PROBABLE RISER GPM	ALLOWABLE DROP (LB PER 100 FT)	PIPE SIZE IN.
				Riser No. 1				
8th	4 W. C. 2 U. 6 3 Lav.	50 40 3	200 80 280 9	40 80	112 7	119	8.5	2½
7th	4 W. C. 2 U. 12 3 Lav. 6	50 40 3	200 80 560 9 18	25 55	140 10	150	8.5	2½
6th	4 W. C. 2 U. 18 3 Lav. 9	50 40 3	200 80 840 9 27	18 50	151 14	165	8.5	2½
5th	4 W. C. 2 U. 24 3 Lav. 12	50 40 3	200 80 1120 9 36	15 47	168 17	185	8.5	3
4th	24 W. C.[a] and U. 10 Lav. 22	3	1120 30 66	15 40	168 27	195	8.5	3
3rd	24 W. C.[a] and U. 1 S. S. 23	4	1120 4 70	15 40	168 28	196	8.5	3
2nd	24 W. C.[a] and U. 1 S. S. 24	4	1120 4 74	15 40	168 30	198	8.5	3
1st	24 W. C.[a] and U. 1 S. S. 25	4	1120 4 78	15 40	168 31	199	8.5	3

[a]From floors above.

supply fixtures must have these returns carried down to the basement from the termination of the supply riser at whatever level it may end.

All risers, both hot and cold, should be valved at the main with an extra check valve on the hot water return circulation so that the risers may be cut off and repaired when necessary without disturbing the service in the remainder of the system.

HOT WATER SUPPLY

Having designed the service hot water piping, the next step is to furnish

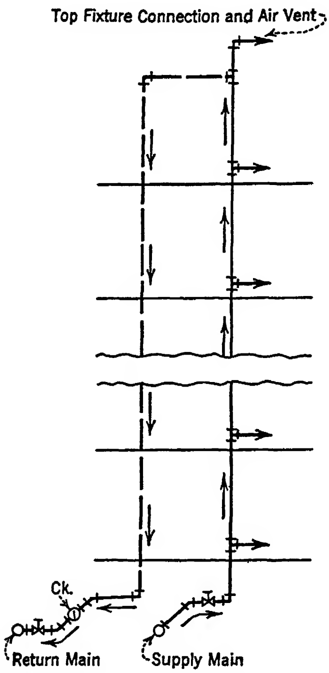


FIG. 7. SUPPLY AND RETURN MAIN CONNECTIONS FOR HOT WATER SUPPLY SYSTEM

some means of heating the water and in this respect it is necessary to pass from the maximum probable flow to the *maximum probable hourly demand*, which is quite different. If an instantaneous heater were used, it would require adequate capacity to provide for the heating of the water as fast as it is drawn and a heater of this type should be sized on the basis of the maximum probable flow with the accompanying heavy drafts on the heating device and with intervals of no draft at all. To balance these inequalities of flow the storage-type heater is often utilized so that the water demand can be heated during periods of light demand and stored up for use during the periods of heavy demand. The total water consumption per person usually varies between 100 and 150 gal per day when

laundry and culinary operations for the occupants are carried out on the same premises. The maximum hourly demand under these conditions will be found to be about one-tenth of the average daily consumption.

If one-third of the total water used is hot water and 125 gal per day is assumed as a fair average of consumption per person, it is apparent that each person uses about 40 gal of hot water per day. If one-tenth of this represents the peak hourly load, then 4 gph must be allowed per person for the heaviest demand. If the average occupancy of apartments is 3 persons, the peak hour demand per apartment will be about 12 gph. It is customary to allow 10 gph of heating capacity per apartment. Water in excess of this heating capacity drawn out during the peak hours is provided for by storage in the hot water tank where this water is heated during hours when the demand is below the average.

HOT WATER STORAGE

The amount of storage provided in the hot water tank or heater is somewhat a matter of choice but is usually made ample to carry over the peak shortage which is likely to occur and is based on the assumption that only 75 per cent of the storage capacity will be available, as it has been found that if more than this amount is withdrawn from storage, the tank is so cooled down as to make the balance useless. The general rule may be cited that the less the heating capacity the greater must be the storage, and the greater the storage the less may be the heating capacity down to a point where the heating capacity will fail to be sufficient to heat up the tank storage during the periods of small load.

Example 5. A heater to supply 500 persons will have an average daily use of about

$$500 \times 40 \text{ gal or } 20{,}000 \text{ gal}$$

and this is an average of

$$\frac{20{,}000 \text{ gal}}{24} \text{ or } 833 \text{ gph}$$

but the peak hour will require

$$\tfrac{1}{10} \text{ of } 20{,}000 \text{ or } 2{,}000 \text{ gal}$$

and the shortage during the peak hour, if the heating capacity is made to suit the average hourly use of 833 gal, will be

$$2{,}000 - 833 \text{ or } 1167 \text{ gal}$$

so that the storage capacity, based on 75 per cent being available from this capacity without cooling the tank excessively, will be

$$\frac{1167}{0.75} \text{ or } 1556 \text{ gal.}$$

Should it be desired to reduce the size of storage tanks and to use a greater heating capacity, it is only necessary to increase the heating capacity to say 1200 gph which then gives

$$2{,}000 - 1200 \text{ or } 800 \text{ gal}$$

as the shortage during the peak hour, and the necessary storage will be

$$\frac{800 \text{ gal}}{0.75} \text{ or } 1067 \text{ gal;}$$

or the heating capacity can be increased to 1500 gal, leaving a shortage of

$$2000 - 1500 \text{ or } 500 \text{ gal}$$

TABLE 11. ORDINARY MAXIMUM HOURLY DEMAND FOR HOT WATER FOR VARIOUS FIXTURES IN GALLONS AND PROBABLE PERCENTAGE OF USAGE

TYPE OF BUILDING	LAVATORIES		BATHS	SHOWERS	SLOP SINKS	KITCHEN SINKS	PANTRY SINKS	FOOT BATHS	WASH TRAYS	AV. MAX. USE[a]
	Private	Public								
MAXIMUM PROBABLE USAGE GPM	20	20	40	300	30	30	20	20	50	
	Probable Usage in Per Cent of Maximum Ordinary									
Apt. House	25	50	33	67	67	33	50	25	60	35
Club	25	75	50	67	67	67	100	25	80	60
Gym.	25	100	100	100	......	......	......	100	......	80
Hospital	25	75	50	33	67	67	100	25	80	45
Hotel	25	100	50	33	100	67	100	25	80	70
Industrial	25	150	100	100	67	67	......	100	......	90
Laundries	25	100	......	......	33	......	......	......	100	100
Office Bldg.	25	75	......	......	50	......	......	......	......	20
Baths	25	150	150	100	50	......	......	......	......	100
Residences	25	......	50	33	50	33	50	50	60	50
Schools	25	75	......	100	67	33	100	50	......	25
Y. M. C. A.	25	100	100	100	67	67	100	100	80	75

[a]Percentage of fixtures likely to be demanding maximum probable usage at any one time.

TABLE 12. HOT WATER CONSUMPTION IN VARIOUS TYPES OF BUILDINGS FOR DIFFERENT PURPOSES

TYPE OF BUILDING	CONDITIONS	GALLONS
Hotels	Room with basin only	10 (per day)
	Room with bath	
	(Transient)	40 (per day)
	(Men)	40 (per day)
	(Mixed)	60 (per day)
	(Women)	80 (per day)
	Two-room suite and bath	80 (per day)
	Three-room suite and bath	100 (per day)
Public Buildings	Public bath or lavatory	150 (per day per fixture)
	Public shower	200 (per day per fixture)
	Public lavatory with attendant	200 (per day per fixture)
Industrial Buildings	Per office employee	2 (per day)
	Per factory employee	5 (per day)
	Cleaning floors	3 (per 1000 sq ft per day)
Restaurant	$0.50 Meals	0.5 (per customer with hand washing)
		1.0 (per customer with machine washing)
	$1.00 Meals	1.0 (per customer with hand washing)
		2.0 (per customer with machine washing)
	$1.50 Meals	1.5 (per customer with hand washing)
		4.0 (per customer with machine washing)

and the storage required only

$$\frac{500}{0.75} \text{ or } 667 \text{ gal.}$$

Good design requires that the heating capacity be made as small as possible without introducing undesirable amounts of storage as the heating capacity directly determines the load on the source of heat.

As indicated in Example 5, the heating load is proportional to the heating capacity and the boiler capacity must be increased for higher heating capacities and may be reduced for smaller heating capacities with greater storage. It may be assumed that a boiler capacity of about 3½ sq ft[3] of equivalent steam heating surface (radiation) must be provided for every gallon of water heated 100 deg or from 50 F to 150 F, which is the temperature rise most commonly assumed and required. On this basis it will be seen that the various conditions cited in Example 5 will require additional boiler capacity as follows:

Heating Capacity (gph)	Additional Boiler Capacity (Sq Ft EDR)
833	2916
1200	4200
1500	5250

From this it is apparent that it is less costly to provide ample storage and to reduce boiler capacity than to diminish the storage and supply a greatly increased boiler capacity to compensate.

ESTIMATING HOT WATER DEMAND BY FIXTURES

In buildings where the occupancy is doubtful and only the number of plumbing fixtures can serve as a basis for determining the probable hot water demand, the problem is not so simple owing to the fact that a fixture gives no information as to how heavy a service may be demanded from the fixture and this amount of service is really the governing factor in making an estimate of the probable hot water demand. Table 11 may prove of some value in this respect as it gives the maximum assumed quantity of hot water per hour which will be demanded of any fixture and then gives a percentage of this amount which may be assumed as probable in different types of buildings. Table 12 gives approximate hot water requirements in various types of buildings.

Example 6. Let it be assumed that an apartment house with 20 apartments has 20 baths, 20 lavatories, 20 kitchen sinks and 20 laundry trays; what is the probable maximum hourly demand for hot water?

20 Baths at 40 gal and 33 per cent	270 gal
20 Lavs. at 20 gal and 25 per cent	100 gal
20 Sinks at 30 gal and 33 per cent	200 gal
20 Trays at 50 gal and 60 per cent	600 gal
Total	1170 gal
Probable peak use at one time	35 per cent
Probable actual peak demand	409 gph

[3]Actual requirement for 100-deg temperature difference $= \frac{100 \times 8.33}{240} = 3.33$ sq ft per gallon of water heated.

If three persons are assumed to an apartment the total daily use of hot water should approximate

$$20 \times 3 \times 40 \text{ gal or } 2400 \text{ gal}$$

and if the peak hour is 10 per cent of this amount, the peak hour by this method shows a probable demand of one-tenth of 2400 gal which indicates that the values in Table 7 are safe.

Chapter 40

TEST METHODS AND INSTRUMENTS

Pressure Measurement, Temperature Measurement, Air Movement, Humidity Measurement, Carbon Dioxide Determination, Dust Determination, Flue Gas Analysis, Measurement of Smoke Density, Heat Transmission, Eupatheoscope

PRESSURE MEASUREMENT

ATMOSPHERIC pressure is usually measured by a *mercurial barometer* which, in its simplest form, consists of a glass tube about 3 ft long, closed at the upper end, filled with mercury and inverted in a shallow bath of mercury. The atmosphere, pressing on the exposed top of the mercury in the cistern, supports a column of mercury in the tube to a height of about 30 in. Readings are taken of the height of the column between the levels of mercury in the tube and in the cistern. Atmospheric pressure is the same as the pressure exerted by this supported column of mercury, and, in pounds per square inch, is equal to its height in inches times 0.491, which is the weight in pounds of 1 cu in. of mercury. At latitude 45 deg and sea level, and at a temperature of 32 F, the atmosphere will support a column of mercury 29.921 in. in height. The pressure of 14.7 lb per square inch, derived by multiplying 29.921 by 0.491, is called *standard* or *normal barometric pressure*. An *aneroid barometer* contains no liquid. Atmospheric pressure in bending the thin corrugated top of a partially exhausted metallic box, or in distorting a thin-walled bent tube of metal, is made to move a pointer. It is portable but less accurate than the mercurial barometer.

Pressures above or below atmospheric are usually measured by means of gages which indicate the difference between the pressure being measured and atmospheric pressure at the same time and place. A gage which indicates pressures higher than atmospheric is known as a *pressure gage*, and a gage which indicates pressures lower than atmospheric is known as a *vacuum gage*. The most common type of these gages contains a flexible hollow brass tube of oval cross section, known as a *Bourdon tube*. When subjected to unequal inside and outside pressures, this tube tends to straighten out, and a pointer motivated by this straightening indicates the pressure difference on a suitably graduated scale.

A gage which indicates pressures slightly above or below atmospheric is known as a *draft gage*. It is essentially a *U* tube containing either water, kerosene, alcohol or mercury, with one leg exposed to the air and the other connected to a point where the pressure is to be determined. When the pressure being read is equal to atmospheric, the level of the liquid in the legs will be the same, indicating a zero gage pressure. When a pres-

sure is applied to one leg, one side will fall and the other will rise an equal amount. The difference in height between the two liquid levels indicates the pressure expressed in inches of liquid used in the gage.

TEMPERATURE MEASUREMENT

In engineering work, *mercurial thermometers* are largely employed to measure the intensity of heat. These depend on the uniform expansion of mercury to indicate changes in temperature. An amount of mercury held in a sealed tube with a bulb at one end will rise to one definite level when immersed in melting ice, and to another definite level when immersed in boiling water. These two points are marked, and the space between them is divided into a number of equal portions, each of which is called a degree. In the Fahrenheit scale, there are 180 deg thus obtained, while the centigrade scale has 100 and the Réaumur has 80. Like divisions are marked off on the column above and below these two determined points in order that a greater range of temperature may be read.

Thermocouples[1] may be used to measure any range of temperatures up to 2,900 F. When two dissimilar metals are joined at two points and a temperature difference exists between these junctions, an electromotive force will be developed. Its magnitude depends on the composition of the wires and the difference in temperature between the junctions. A potentiometer or sensitive galvanometer of high resistance connected to the thermocouple will give a deflection which is proportional to the temperature difference between the hot and cold junctions. Thermocouples connected in series are called *thermopiles*. Thermocouples for the measurement of high temperatures are calibrated with the aid of the known melting points of pure metals.

For temperatures above 500 F various types of *pyrometers* are employed. The *mercurial pyrometer* is a thermometer with an inert gas, such as nitrogen or carbon dioxide, above the mercury column to prevent the mercury from boiling. The *radiation pyrometer* consists of a thermopile upon which the radiation from a hot source is focused by a concave mirror. A sensitive galvanometer with a calibrated temperature scale indicates the thermo-electromotive force created by the heat on the thermopile. The *optical pyrometer* measures radiant energy by comparing the intensity of a narrow spectral band, usually red light emitted by the object, with that emitted by a standard light source (electric lamp). *Thermo-electric pyrometers* operate on the same principle as thermocouples. When measuring high temperatures, it is customary to hold the cold junction at room temperature which may cause some error if the room temperature is above or below the calibration point.

In the measuring of room temperatures care must be exercised to prevent the results from being affected by the body heat of the observer, by drafts from doors, windows and other openings, or by radiant heat from some local source such as a radiator or wall. All thermometers should be mercury thermometers with engraved stems. The total graduations of the thermometers should be from 20 to 120 F, in one degree

[1]See A.S.H.V.E. research paper entitled Study of the Application of Thermocouples to the Measurement of Wall Surface Temperatures, by A. P. Kratz and E. L. Broderick (A.S.H.V.E. Transactions, Vol. 38, 1932).

graduations. No ten degrees should occupy a space of less than one-half inch. The accuracy throughout the whole scale must be within one-half degree. The operator should take hold of the top and no part of the body, including the hand, should be nearer than 10 in. to the bulb. The thermometer should not be closer than 5 ft to any door, window, or other opening; should not be closer than 12 in. to any wall; and should be between 3 and 5 ft from the floor. A sling instrument should be used for extreme accuracy.

For measuring duct temperatures an angle-duct thermometer should be used, having a flange to bolt on the side of the duct, with the bulb extending into the duct at least 6 in.

Recording thermometers generally have considerable lag and should not be used for the taking of temperatures for testing, but rather for giving continuous records of the operation of the plant, as the charts will indicate any lack of attention on the part of those responsible for the operation of the plant.

MEASUREMENT OF AIR MOVEMENT

The quantity, velocity and pressure of air discharged by a fan or flowing through a duct or grille may be determined by various methods. Those in common use are by Pitot tube, anemometer and Kata-thermometer readings, the latter being suitable for low air velocities and being commonly used for measurements at points where the air is not confined in a duct. The use of calibrated nozzles, orifice plates, and Venturi meters are recognized methods, which, however, have little application in connection with ventilation practice.

Pitot Tube

This usually consists of two tubes, one within the other, which when properly held in the air stream will register the total or impact pressure and the static pressure, respectively. If these tubes are connected to opposite sides of a water column the recorded pressure will be the differential or velocity head. Volume measurements may thus be made in a duct of known area. Pitot tube measurements are preferably used for air velocities exceeding 20 fps. Volumetric determinations from Pitot tube readings must take into account the barometric pressure, temperature and humidity. These factors determine the weight or density of the air.

In general no accurate velocity pressure readings can be taken when the flow of air in ducts is turbulent. To insure accuracy a straight section of duct from 5 to 10 times its own diameter is desirable in order to straighten out the air currents. If it is necessary to take Pitot tube readings in shorter sections of straight duct, the results must be considered subject to some doubt and checked accordingly. For accurate work it is necessary to make a traverse of the duct, dividing its cross section into a number of imaginary equal areas and taking a reading in the center of each, the average of the velocities corresponding to these pressures giving the true velocity in the duct.

Anemometer

This instrument is delicate, and requires frequent calibration when

accuracy is desired. In duct measurements the same procedure is followed as for the Pitot tube. The anemometer usually reads directly in linear feet. To obtain the velocity in feet per minute, the reading must be divided by the elapsed time in minutes.

The following procedure for obtaining anemometer readings is based on research conducted at *Armour Institute of Technology* in coöperation with the A.S.H.V.E. Research Laboratory[2].

Supply Grilles. The surface of the grille should be marked off into a number of equal areas approximately 6 in. square. A 4-in. anemometer should be used and should be held at the center of each section in contact with the grille (or as close as possible) for a period of time sufficient to insure an average reading. In the case of supply grilles, the instrument should always be held with the dial facing the operator. The average of the corrected readings should then be used in the following formula to obtain the flow in cubic feet per minute:

$$cfm = CV\frac{A + a}{2} \text{ or } \frac{CVA\,(1 + p)}{2} \tag{1}$$

where

V = average of corrected anemometer readings in feet per minute.
A = gross area of grille, square feet.
a = net free area of grille, square feet.
p = percentage of free area of grille expressed as a decimal.
C = a coefficient that varies with the velocity from grille and may vary slightly with type of grille. For average use, with supply grilles, C can be taken as 0.97 at velocities from 150 to 600 fpm, and as 1.00 at higher velocities.

Particular care should be exercised in the case of long, narrow grilles. The nature of the approach sometimes results in a narrow strip along the top or bottom of the grille through which no air will be flowing. This may be detected by holding the anemometer completely out of the air stream and then moving it slowly inward over the grille until the vanes just start to move. The distance which the vanes extend over the grille opening at this moment will indicate the width of the dead strip. Only the remaining portion of the grille should be considered in making the calculations for gross and free area.

Exhaust Grilles. The surface of the grille should be marked off and readings taken in the same manner as with supply grilles, except that the instrument should be held with the dial facing the grille, and in contact with it. The traverse should be taken at a uniform rate, allowing sufficient time in each space to minimize the percentage of error. In the case of exhaust grilles it is found that the formula

$$cfm = KVA \tag{2}$$

in which

V = average indicated velocity obtained by the anemometer traverse in contact with grille.

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[2]Measurement of Flow of Air through Registers and Grilles, by L. E. Davies (A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, Vol. 37, 1931, and A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, September, 1933).

A = gross area of grille, square feet.
K = coefficient determined by experiment. For average use, with exhaust grilles, K may be taken as 0.8 for all usual velocities.

This formula is of advantage, especially with ornamental grilles, in that the free area need not be measured.

The flow of air through registers and grilles is of considerable importance, being frequently the only convenient method of measuring the volume of supply air to a room. While duct measurements, if available, are more dependable, grille measurements provide a fairly accurate method, if care is taken in the technique of using the anemometer.

Kata-Thermometer

The Kata-thermometer can be used as an anemometer provided the walls and surrounding objects are at or near the room temperature. Especially at low velocities it constitutes a useful instrument for readily detecting drafts.

The instrument is essentially an alcohol thermometer with a bulb approximately ⅝ in. in diameter and ½ in. long with a stem 8 in. long reading from 100 to 95 F, graduated to tenths of a degree. To take readings the bulb is heated in water until the alcohol expands and rises into a top reservoir. The time in seconds required for the liquid to fall from 100 F to 95 F is recorded with a stop watch and this time is a measure of the rate of cooling.

A dry Kata gives the cooling power by radiation and convection. A wet Kata, which has a cotton lisle wick fitted snugly around the bulb, gives the cooling power by radiation, convection and evaporation. For constant velocities the time of cooling of the dry Kata is a function of the dry-bulb temperature alone, while that of the wet Kata is a function of the wet-bulb temperature regardless of the dry-bulb temperature or the relative humidity. Due to the comparatively brief time of fall of the wet Kata-thermometer, the dry Kata-thermometer is far more accurate for measuring air motion since any probable error in recording the time of fall will only amount to a small fraction of the total period.

HUMIDITY MEASUREMENT

The sling psychrometer is the recognized standard instrument for determining humidities. In order to obtain accurate readings considerable skill is required on the part of the operator. The wicking must be clean, distilled water should be used, and the temperature of the water should be slightly above the wet-bulb temperature of the surrounding air. The psychrometer should be swung rapidly and two or three observations should be made to see that the wet-bulb temperature has become stationary before the final reading is noted. Standard psychrometric tables should be used.

In making wet-bulb measurements below 32 F the same procedure is followed as above 32 F. The water is liquid at the start, but as the sling is operated it will freeze rapidly enough so that in quickly giving up the latent heat of fusion, the indicated wet-bulb temperature may drop below the actual wet-bulb temperature. After the liquid on the bulb has

become thoroughly frozen the wet-bulb temperature will rise to normal. Care must be taken to read the temperatures in the region below 32 F accurately because the spread between the wet- and dry-bulb is small.

In taking humidity readings in ducts it is usually impracticable to use a sling psychrometer. For this work the stationary hygrodeik arranged for bolting on to the side of the duct, with two bulbs extending into the duct, will be found very convenient. Due to the velocity of the air passing over the bulbs within the duct an accurate reading will be secured, corresponding to that given by the sling psychrometer.

CARBON DIOXIDE DETERMINATION[3]

At ordinary concentrations carbon dioxide is not harmful. The amount of carbon dioxide in the air is a convenient index of the rate of air supply, and of the distribution of the air within rooms. A high carbon dioxide concentration in parts of an occupied room may indicate air stagnation which can result in objectionable odors or in failure to remove local surplus heat.

The Petterson-Palmquist apparatus has been generally accepted as the standard device for the determination of carbon dioxide in air investigations. The principle involved is the measurement of a given volume of air, the absorption of the contained carbon dioxide in a caustic potash solution, and the remeasurement of the volume of air at the original pressure in a finely graduated capillary tube, the difference in volume representing the absorbed carbon dioxide. (See Report of Committee on Standard Methods for Examination of Air, *American Public Health Association*, Vol. 7, No. 1; *American Journal of Public Health*, Jan., 1917).

Where field conditions are such that this apparatus may not be conveniently used, as in street cars, air samples may be collected in prepared bottles having rubber stoppers, and these may be subjected to laboratory analysis.

DUST DETERMINATION

Many laboratory methods have been developed to measure the dust in the air. These involve the collection of dust on sticky plates, on filter paper, in water, on porous crucibles, or by electric precipitation, and the subsequent determination of the amount of dust by microscopic counting, weighing, or titration. While there is no standard method, the Hill dust counter, using a microscope, the impinger[4], using chemical changes in water, and the Lewis sampling tube[5], involving the analytical weighing of a porous crucible, are accepted.

FLUE GAS ANALYSIS

The analysis of flue gases by chemical means is made with the *Orsat apparatus*. A solution of *KOH* is used to absorb the CO_2. Free oxygen is

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[3]See A.S.H.V.E. research paper entitled Indices of Air Change and Air Distribution, by F. C. Houghten and J. L. Blackshaw (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, June, 1933).

[4]*Public Health Bulletin, No.* 144, 1925, U. S. Public Health Service.

[5]Testing and Rating of Air Cleaning Devices Used for General Ventilation Work, by Samuel R. Lewis (A.S.H.V.E. Journal Section, *Heating, Piping and Air Conditioning*, May, 1933).

absorbed by a mixture of pyrogallic acid and KOH. The solution for absorbing the CO is cuprous chloride. The apparatus consists of a burette surrounded by a water jacket, to receive and measure the volume of gas. The burette is connected by a manifold of glass to *pipettes* containing liquids for absorbing CO_2, O_2 and CO.

MEASUREMENT OF SMOKE DENSITY

Smoke density is usually measured by comparison with the Ringelmann Chart (Fig. 1). In making observations of the smoke issuing from a chimney, four cards ruled like those in Fig. 1, together with a card printed in solid black and another left entirely white, are placed in a horizontal row and hung at a point 50 ft from the observer and as nearly as convenient in line with the chimney. At this distance, the lines become invisible, and the cards appear to be of different shades of gray,

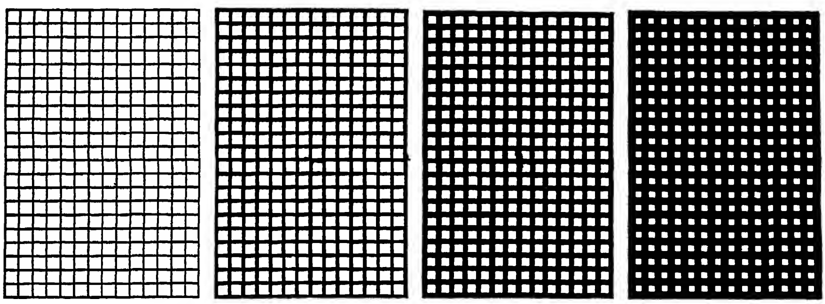

FIG. 1. RINGELMANN SMOKE CHART

ranging from very light gray to almost black. The observer glances from the smoke coming from the chimney to the cards, which are numbered from 0 to 5, determines which card most nearly corresponds with the color of the smoke, and makes a record accordingly, noting the time. Observations are made continuously during say one minute, and the estimated average density during that minute recorded, and so on, records being made once every minute. The average of all the records made during a boiler test is taken as the average figure for the smoke density during the test, and the entire record is plotted on cross section paper in order to show how the smoke varied in density from time to time.

Smoke Recorders

Smoke recorders are available which give a much more accurate indication of the amount of smoke being produced than the Ringelmann Chart. Although most of these instruments are in the process of development, they constitute a satisfactory tool in the control of smoke emission. They all depend upon projecting a beam of light through the smoke flue or through a separate compartment from which a sample of the flue gas is drawn continuously. The light of the beam which passes through without being absorbed by the smoke is measured to determine the smoke

density. Most of these instruments make use of a photo-electric cell or a thermopile to measure the relative amount of light which has not been absorbed. Standard electrical instruments serve for indicating or recording.

MEASUREMENT OF RATE OF HEAT TRANSMISSION

The standard methods of testing built-up wall sections are by means of the *guarded hot-box*[6] and the *guarded hot-plate*[6]. The *Nicholls heat-flow meter*[7] may be used for testing actual walls of buildings.

It would be obviously impossible to determine the air-to-air heat transmission coefficients of every type of wall construction in use with the heat-flow meter, the guarded hot-box or the guarded hot-plate on account of the great amount of time involved. Hence, the method of computing the coefficients from the fundamental constants must be resorted to in most cases. The guarded hot-plate is used to determine the fundamental constants. The heat-flow meter, guarded hot-box and guarded hot-plate tests can be used to good advantage in checking the accuracy of the computed values.

If the hot-box or hot-plate methods are used, tests are usually run under still air conditions, which means there is no wind movement over the surfaces of the wall during the test. In the hot-plate method of test the inside surface coefficient is eliminated by the plate's being in direct contact with the wall. In practice, some wind movement over the exterior surface of the wall should always be allowed for; hence, still-air coefficients cannot be used over the outside of the building during the heating season. Moreover, still-air transmission coefficients cannot be corrected to provide for moving-air conditions by applying a single constant factor. Computed coefficients of transmission for various types of construction are given in Chapter 5.

EUPATHEOSCOPE

The eupatheoscope affords a means of evaluating the combined effect of radiation and convection in a given environment in terms of a standard environment and in some terms related to human comfort. See Chapter 37.

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[6]See Standard Code for Heat Transmission through Walls (A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928) and Report of the Committee on Heat Transmission, *National Research Council.*

[7]See Measuring Heat Transmission in Building Structures and a Heat Transmission Meter, by P. Nicholls (A.S.H.V.E. TRANSACTIONS, Vol. 30, 1924).

Chapter 41

PROPERTIES OF AIR, WATER, AND STEAM

Composition of Air, Specific Density, Boyle's Law, Charles' Law, Composition of Water, Density, Water Pressures, Boiling Point, Specific Heat, Sensible and Latent Heat, Saturated Steam, Superheated Steam, Quality, Tables

AIR is a mechanical mixture made up, by volume, of 20.91 per cent of oxygen and 79.09 per cent of nitrogen, or by weight, of 23.15 per cent of oxygen and 76.85 per cent of nitrogen. Air as found in nature always contains other constituents in varying amounts, such as carbon dioxide, ozone, water vapor, dust, and bacteria.

The *specific density* or weight per cubic foot of dry air decreases with an increase in temperature; and the specific volume, or volume per pound, increases with such increase. The *specific heat* of air at constant pressure, or the Btu required to raise the temperature of one pound one degree Fahrenheit, varies from 0.2375 to 0.2430 as determined by various investigators. The value 0.24 is recommended for engineering calculations. It has been found that a given volume of air expands when heated under constant pressure, and again that if the temperature of a given volume of air be kept constant and the pressure increased, contraction takes place. These changes follow definite laws, which apply to other gases as well as air, known as the *laws of perfect gases.* These laws do not generally apply to steam, since it is not a perfect gas, but superheated steam at high temperatures follows the laws approximately. (See Tables 1, 2 and 3).

Example 1. To show the use of table on air weights, Table 3. Given air at 83 F dry-bulb and 68 F wet-bulb (or a depression of 15 deg) with a barometric pressure of 29.40 in. of mercury. What will be the weight of this air in pounds per cubic foot?

Solution. From Table 3 the weight of saturated air at 80 F and 29.00 in. barometer is found to be 0.07034 lb per cubic foot. There is a decrease of 0.00015 lb per degree dry-bulb temperature above 80 F. There is an increase of 0.00025 lb for each 0.1 in. above 29.00 in. From the last column of Table 3 it is found that there is an increase of approximately 0.000035 lb per degree wet-bulb depression when the dry-bulb is 83 F. Tabulating the items:

0.07034	=	weight of saturated air at 80 F and 29.00 bar.
− 0.00045	=	decrement for 3 deg dry-bulb, 3 × 0.00015.
+ 0.00100	=	increment for 0.4 in. bar, 4 × 0.00025.
+ 0.00053	=	increment for 15 deg wet-bulb depression, 15 × 0.000035.
0.07142	=	weight in pounds per cubic foot of air at 83 F dry-bulb, 68 F wet-bulb, 29.40 in. bar.

Boyle's Law refers to the relation between the pressure and volume of a gas, and may be stated as follows: *With temperature constant, the volume of*

TABLE 1. PROPERTIES OF DRY AIR[a]

Barometric Pressure 29.921 In.

Temperature Deg F	Weight per Cu Ft Pounds	Per Cent of Volume at 70 F	Btu Absorbed by One Cu Ft Dry Air per Deg F	Cu Ft Dry Air Warmed One Degree per Btu
0	0.08636	0.8680	0.02080	48.08
10	0.08453	0.8867	0.02039	49.05
20	0.08276	0.9057	0.01998	50.05
30	0.08107	0.9246	0.01957	51.10
40	0.07945	0.9434	0.01919	52.11
50	0.07788	0.9624	0.01881	53.17
60	0.07640	0.9811	0.01846	54.18
70	0.07495	1.0000	0.01812	55.19
80	0.07356	1.0190	0.01779	56.21
90	0.07222	1.0380	0.01747	57.25
100	0.07093	1.0570	0.01716	58.28
110	0.06968	1.0756	0.01687	59.28
120	0.06848	1.0945	0.01659	60.28
130	0.06732	1.1133	0.01631	61.32
140	0.06620	1.1320	0.01605	62.31
150	0.06510	1.1512	0.01578	63.37
160	0.06406	1.1700	0.01554	64.35
180	0.06205	1.2080	0.01506	66.40
200	0.06018	1.2455	0.01462	68.41
220	0.05840	1.2833	0.01419	70.48
240	0.05673	1.3212	0.01380	72.46
260	0.05516	1.3590	0.01343	74.46
280	0.05367	1.3967	0.01308	76.46
300	0.05225	1.4345	0.01274	78.50
350	0.04903	1.5288	0.01197	83.55
400	0.04618	1.6230	0.01130	88.50
450	0.04368	1.7177	0.01070	93.46
500	0.04138	1.8113	0.01018	98.24
550	0.03932	1.9060	0.00967	103.42
600	0.03746	2.0010	0.00923	108.35
700	0.03423	2.1900	0.00847	118.07
800	0.03151	2.3785	0.00782	127.88
900	0.02920	2.5670	0.00728	137.37
1000	0.02720	2.7560	0.00680	147.07

[a]From *Fan Engineering*.

a given weight of gas varies inversely as its absolute pressure. Hence, if P_1 and P_2 represent the initial and final absolute pressures, and V_1 and V_2 represent corresponding volumes of the same mass, say one pound of gas, then $\frac{V_1}{V_2} = \frac{P_2}{P_1}$, or $P_1 V_1 = P_2 V_2$, but since $P_1 V_1$ for any given case is a definite constant quantity, it follows that the product of the absolute pressure and volume of a gas is a constant, or $PV = C$, when T is kept

TABLE 2. PROPERTIES OF SATURATED AIR[a]

Weights of Air, Vapor of Water, and Saturated Mixture of Air and Vapor at 29.921 Inches of Mercury

TEMP. DEG F	WEIGHT IN A CUBIC FOOT OF MIXTURE			BTU ABSORBED BY ONE CUBIC FOOT SAT. AIR PER DEG F	CUBIC FEET SAT. AIR WARMED ONE DEGREE PER BTU	SPECIFIC HEAT BTU PER POUND OF MIXTURE
	WEIGHT OF DRY AIR POUNDS	WEIGHT OF VAPOR POUNDS	TOTAL WEIGHT OF THE MIXTURE POUNDS			
0	0.08625	0.000068	0.08632	0.02083	48.02	0.2413
10	0.08433	0.000110	0.08444	0.02039	49.05	0.2415
20	0.08246	0.000176	0.08264	0.01998	50.07	0.2418
30	0.08062	0.000277	0.08090	0.01958	51.07	0.2420
40	0.07878	0.000409	0.07919	0.01921	52.06	0.2426
50	0.07694	0.000587	0.07753	0.01885	53.05	0.2431
60	0.07506	0.000828	0.07589	0.01851	54.02	0.2439
70	0.07310	0.001151	0.07425	0.01819	54.97	0.2450
80	0.07103	0.001578	0.07261	0.01790	55.87	0.2465
90	0.06879	0.002134	0.07092	0.01762	56.76	0.2485
100	0.06635	0.002850	0.06920	0.01736	57.59	0.2509
110	0.06364	0.003762	0.06740	0.01714	58.35	0.2543
120	0.06060	0.004914	0.06551	0.01695	59.00	0.2587
130	0.05715	0.006351	0.06350	0.01679	59.56	0.2644
140	0.05319	0.008120	0.06131	0.01668	59.96	0.2721
150	0.04864	0.010295	0.05894	0.01662	60.17	0.2820
160	0.04340	0.012936	0.05634	0.01662	60.17	0.2950
170	0.03734	0.016108	0.05345	0.01668	59.96	0.3121
180	0.03035	0.019896	0.05025	0.01684	59.38	0.3351
190	0.02228	0.024400	0.04668	0.01710	58.49	0.3663
200	0.01300	0.029715	0.04272	0.01749	57.18	0.4094
210	0.00230	0.035938	0.03824	0.01802	55.50	0.4712
212	0.00000	0.037307	0.03731	0.01815	55.10	0.4865

[a]From *Fan Engineering*.

constant. Any change in the pressure and volume of a gas at constant temperature is called an *isothermal change.*

Charles' Law refers to the relation between pressure, volume and temperature of a gas and may be stated as follows: *The volume of a given weight of gas varies directly as the absolute temperature at constant pressure, and the pressure varies directly as the absolute temperature at constant volume.* Hence, when heat is added at constant volume, V_c, the resulting

equation is $\frac{P_2}{P_1} = \frac{T_2}{T_1}$, or, for the same temperature range at constant pressure, P_c, the relation is $\frac{V_2}{V_1} = \frac{T_2}{T_1}$.

In general, for any weight of gas, W, since volume is proportional to weight, the relation between P, V and T is

$$PV = WRT \tag{1}$$

where

P = the absolute pressure of the gas in pounds per square foot.
V = the volume of the weight W in cubic feet.
W = the weight of the gas in pounds.
R = a constant depending on the nature of the gas.
T = the absolute temperature in degrees Fahrenheit.

This is the characteristic equation for a perfect gas, and while no gases are perfect in this sense, they conform so nearly that Equation 1 will apply to most engineering computations.

Additional information on the properties of air will be found in Chapters 1 and 2.

PROPERTIES OF WATER

Composition of Water. Water is a chemical compound (H_2O) formed by the union of two volumes of hydrogen and one volume of oxygen, or two parts by weight of hydrogen and 16 parts by weight of oxygen.

Density of Water. Water is at its greatest density at 39.2 F, and it expands when heated or cooled from this temperature. At 62 F a U. S. gallon of 231 cu in. of water weighs approximately 8⅓ lb, and a cubic foot of water is equal to 7.48 gal. The specific volume of water depends on the temperature and it is always the reciprocal of its specific density. (See Table 4).

Water Pressures. Pressures are often stated in feet or inches of water column. At 62 F, with h equal to the head in feet, the pressure of a column of water is $62.383h$ lb per square foot, or $0.433h$ lb per square inch. A column of water 2.309 ft (27.71 in.) high exerts a pressure of one pound per square inch at 62 F.

Boiling Point of Water. The boiling point of water varies with the pressure; it is lower at higher altitudes. A change in pressure will always be accompanied by a change in the boiling point, and there will be a corresponding change in the latent heat of evaporation.

Specific Heat. The specific heat of water, or the amount of heat (Btu) required to raise the temperature of one pound of water one degree Fahrenheit, varies with the temperature, but it is commonly assumed to be unity at all temperatures. Steam tables are based on exact values, however. The specific heat of ice at 32 F is 0.492 Btu per pound. The amount of heat required to raise one pound of water at 32 F through a known temperature interval depends on the average specific heat for the temperature range.

Sensible and Latent Heat. The heat necessary to raise the temperature of one pound of water from 32 F to the boiling point is known as the *heat*

TABLE 3. WEIGHTS OF SATURATED AND PARTLY SATURATED AIR FOR VARIOUS BAROMETRIC AND HYGROMETRIC CONDITIONS[a,b]

Pounds per Cubic Foot

Dry-Bulb Temperature Deg F	Barometric Pressure—Inches: 26			27			28			29			30			Approx. Average Increase in Weight per Deg Wet-Bulb Depression
	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar	Wt. per Cu Ft Saturated Air	Decr's Wt. per Deg Inc. Dry-Bulb	Incr's Wt. per 0.1″ Rise in Bar	
0	0.07500	0.00016	0.00029	0.07788	0.00016	0.00029	0.08077	0.00017	0.00029	0.08365	0.00018	0.00029	0.08654	0.00019	0.00029	0.000015
10	0.07338	0.00016	0.00028	0.07620	0.00016	0.00028	0.07903	0.00017	0.00028	0.08185	0.00018	0.00028	0.08468	0.00018	0.00028	0.000016
20	0.07180	0.00016	0.00028	0.07456	0.00016	0.00028	0.07733	0.00017	0.00028	0.08009	0.00018	0.00028	0.08286	0.00018	0.00028	0.000017
30	0.07027	0.00015	0.00027	0.07297	0.00016	0.00027	0.07569	0.00016	0.00027	0.07839	0.00017	0.00027	0.08110	0.00017	0.00027	0.000019
40	0.06879	0.00015	0.00026	0.07143	0.00015	0.00027	0.07409	0.00016	0.00027	0.07675	0.00016	0.00027	0.07942	0.00017	0.00027	0.000021
50	0.06732	0.00015	0.00026	0.06992	0.00015	0.00026	0.07252	0.00016	0.00026	0.07512	0.00016	0.00026	0.07773	0.00016	0.00026	0.000023
60	0.06588	0.00015	0.00026	0.06843	0.00015	0.00026	0.07098	0.00015	0.00026	0.07353	0.00016	0.00026	0.07609	0.00016	0.00026	0.000026
70	0.06442	0.00015	0.00025	0.06692	0.00015	0.00025	0.06943	0.00015	0.00025	0.07193	0.00016	0.00025	0.07440	0.00016	0.00025	0.000029
80	0.06297	0.00015	0.00025	0.06542	0.00015	0.00025	0.06789	0.00015	0.00025	0.07034	0.00015	0.00025	0.07280	0.00016	0.00025	0.000034
90	0.06146	0.00015	0.00024	0.06388	0.00016	0.00024	0.06629	0.00016	0.00024	0.06870	0.00016	0.00024	0.07112	0.00017	0.00024	0.000039
100	0.05991	0.00016	0.00024	0.06228	0.00016	0.00024	0.06465	0.00016	0.00024	0.06703	0.00017	0.00024	0.06939	0.00018	0.00024	0.000044
110	0.05828	0.00016	0.00023	0.06060	0.00017	0.00023	0.06293	0.00017	0.00023	0.06526	0.00018	0.00023	0.06759	0.00019	0.00023	0.000051
120	0.05653	0.00018	0.00023	0.05882	0.00018	0.00023	0.06111	0.00018	0.00023	0.06339	0.00019	0.00023	0.06569	0.00020	0.00023	0.000059
130	0.05467	0.00019	0.00023	0.05692	0.00019	0.00023	0.05917	0.00019	0.00023	0.06142	0.00020	0.00023	0.06367	0.00022	0.00023	0.000068
140	0.05262	0.00021	0.00022	0.05483	0.00021	0.00022	0.05704	0.00021	0.00022	0.05925	0.00022	0.00022	0.06147	0.00024	0.00022	0.000078
150	0.05036	0.00023	0.00022	0.05253	0.00023	0.00022	0.05471	0.00023	0.00022	0.05689	0.00024	0.00022	0.05906	0.00026	0.00022	0.000090
160	0.04788	0.00025	0.00022	0.05001	0.00025	0.00022	0.05216	0.00026	0.00021	0.05430	0.00026	0.00021	0.05644	0.00029	0.00021	0.000103
170	0.04509	0.00028	0.00021	0.04720	0.00028	0.00021	0.04931	0.00029	0.00021	0.05141	0.00031	0.00021	0.05352	0.00033	0.00021	0.000118
180	0.04197	0.00031	0.00021	0.04404	0.00031	0.00021	0.04611	0.00032	0.00021	0.04818	0.00034	0.00021	0.05026	0.00036	0.00021	0.000134
190	0.03845	0.00035	0.00021	0.04049	0.00036	0.00021	0.04253	0.00036	0.00021	0.04457	0.00037	0.00021	0.04662	0.00038	0.00021	0.000153
200	0.03449	0.00040	0.00020	0.03650	0.00040	0.00020	0.03851	0.00040	0.00020	0.04052	0.00041	0.00020	0.04254	0.00041	0.00020	0.000173

[a]From *Fan Engineering*.

[b]A convenient and accurate chart for quickly determining the weight of air under any condition of dry-bulb, wet-bulb and pressure is *A Chart for Determining the Weight of Moist Air in Pounds per Cubic Foot*, by John E. Younger. Published in *Mechanical Engineering*, June, 1925.

of the liquid or *sensible heat.* When more heat is added, the water begins to evaporate and expand at constant temperature until the water is entirely changed into steam. The heat thus added is known as the *latent heat of evaporation.*

TABLE 4. THERMAL PROPERTIES OF WATER

Temperature Deg F	Sat. Press. Lb per Sq In.	Volume Cu Ft per Lb	Weight Lb per Cu Ft	Specific Heat
32	0.0887	0.01602	62.42	1.0093
40	0.1217	0.01602	62.42	1.0048
50	0.1780	0.01602	62.42	1.0015
60	0.2561	0.01603	62.38	0.9995
70	0.3628	0.01605	62.31	0.9982
80	0.5067	0.01607	62.23	0.9975
90	0.6980	0.01610	62.11	0.9971
100	0.9487	0.01613	62.00	0.9970
110	1.274	0.01616	61.88	0.9971
120	1.692	0.01620	61.73	0.9974
130	2.221	0.01625	61.54	0.9978
140	2.887	0.01629	61.39	0.9984
150	3.716	0.01634	61.20	0.9990
160	4.739	0.01639	61.01	0.9998
170	5.990	0.01645	60.79	1.0007
180	7.510	0.01650	60.61	1.0017
190	9.336	0.01656	60.39	1.0028
200	11.525	0.01663	60.13	1.0039
210	14.123	0.01669	59.92	1.0052
212	14.696	0.01670	59.88	1.0055
220	17.188	0.01676	59.66	1.0068
240	24.97	0.01690	59.17	1.0104
260	35.43	0.01706	58.62	1.0148
280	49.20	0.01723	58.04	1.020
300	67.01	0.01742	57.41	1.026
350	134.62	0.01797	55.65	1.044
400	247.25	0.01865	53.62	1.067
450	422.61	0.0195	51.3	1.095
500	681.09	0.0205	48.8	1.130
550	1045.4	0.0219	45.7	1.200
600	1544.6	0.0241	41.5	1.362
700	3096.4	0.0394	25.4	....

PROPERTIES OF STEAM

Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. This change in state takes place at a definite and constant temperature which is determined solely by the pressure of the steam. The volume of a pound of steam is the *specific volume* which decreases as the pressure increases. The reciprocal of this, or the weight of steam per cubic foot, is the *density*. (See Table 5).

Steam which is in contact with the water from which it was generated is known as *saturated steam.* If it contains no actual water in the form of mist or priming, it is called *dry saturated steam.* If this be heated and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated*, that is, its temperature will

TABLE 5. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE[a]

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
½″ Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	½″ Hg
¾″ Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	¾″ Hg
1″ Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1″ Hg
1½″ Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	1½″ Hg
2″ Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	2″ Hg
2½″ Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	2½″ Hg
3″ Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	3″ Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6055	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

[a]Abstracted from *Steam Tables and Mollier Diagram*, by Prof. J. H. Keenan, 1930 edition, by permission of the publisher, *The American Society of Mechanical Engineers*.

TABLE 5. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Total Heat			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE 5. PROPERTIES OF SATURATED STEAM: PRESSURE TABLE—(Continued)

		Specific Volume			Total Heat			Entropy			
Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Abs. Press. Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
250.0	400.97	0.01867	1.8223	1.8410	376.02	824.5	1200.5	0.5680	0.9581	1.5261	250.0
260.0	404.43	0.01872	1.7536	1.7723	379.78	821.2	1201.0	0.5723	0.9504	1.5227	260.0
270.0	407.79	0.01877	1.6895	1.7083	383.44	818.0	1201.4	0.5765	0.9430	1.5194	270.0
280.0	411.06	0.01882	1.6302	1.6490	387.02	814.7	1201.8	0.5805	0.9357	1.5163	280.0
290.0	414.24	0.01887	1.5745	1.5934	390.50	811.6	1202.1	0.5845	0.9287	1.5132	290.0
300.0	417.33	0.01892	1.5225	1.5414	393.90	808.5	1202.4	0.5883	0.9220	1.5102	300.0
320.0	423.29	0.01901	1.4279	1.4469	400.47	802.5	1203.0	0.5957	0.9089	1.5046	320.0
340.0	428.96	0.01910	1.3439	1.3630	406.75	796.6	1203.4	0.6027	0.8965	1.4992	340.0
360.0	434.39	0.01918	1.2689	1.2881	412.80	790.9	1203.7	0.6094	0.8846	1.4940	360.0
380.0	439.59	0.01927	1.2015	1.2208	418.61	785.3	1203.9	0.6157	0.8733	1.4891	380.0
400.0	444.58	0.0194	1.1407	1.1601	424.2	779.8	1204.1	0.6218	0.8625	1.4843	400.0
420.0	449.38	0.0194	1.0853	1.1047	429.6	774.5	1204.1	0.6277	0.8520	1.4798	420.0
440.0	454.01	0.0195	1.0345	1.0540	434.8	769.3	1204.1	0.6334	0.8420	1.4753	440.0
460.0	458.48	0.0196	0.9881	1.0077	439.9	764.1	1204.0	0.6388	0.8322	1.4711	460.0
480.0	462.80	0.0197	0.9456	0.9633	444.9	759.0	1203.9	0.6441	0.8228	1.4670	480.0
500.0	466.99	0.0198	0.9063	0.9261	449.7	754.0	1203.7	0.6493	0.8137	1.4630	500.0
520.0	471.05	0.0198	0.8701	0.8899	454.4	749.0	1203.5	0.6543	0.8048	1.4591	520.0
540.0	474.99	0.0199	0.8363	0.8562	459.0	744.1	1203.2	0.6592	0.7962	1.4554	540.0
560.0	478.82	0.0200	0.8047	0.8247	463.6	739.3	1202.9	0.6639	0.7878	1.4517	560.0
580.0	482.55	0.0201	0.7751	0.7952	468.0	734.5	1202.5	0.6686	0.7796	1.4482	580.0
600.0	486.17	0.0202	0.7475	0.7677	472.3	729.8	1202.1	0.6731	0.7716	1.4447	600.0
620.0	489.71	0.0202	0.7217	0.7419	476.6	725.1	1201.7	0.6775	0.7638	1.4413	620.0
640.0	493.16	0.0203	0.6972	0.7175	480.8	720.5	1201.2	0.6818	0.7562	1.4380	640.0
660.0	496.53	0.0204	0.6744	0.6948	484.9	715.9	1200.8	0.6861	0.7487	1.4348	660.0
680.0	499.82	0.0205	0.6527	0.6732	488.9	711.3	1200.2	0.6902	0.7414	1.4316	680.0
700.0	503.04	0.0206	0.6321	0.6527	492.9	706.8	1199.7	0.6943	0.7342	1.4285	700.0
720.0	506.19	0.0206	0.6128	0.6334	496.8	702.4	1199.2	0.6983	0.7272	1.4255	720.0
740.0	509.28	0.0207	0.5944	0.6151	500.6	697.9	1198.6	0.7022	0.7203	1.4225	740.0
760.0	512.30	0.0208	0.5769	0.5977	504.4	693.5	1198.0	0.7060	0.7136	1.4196	760.0
780.0	515.27	0.0209	0.5602	0.5811	508.2	689.2	1197.4	0.7098	0.7069	1.4167	780.0
800.0	518.18	0.0209	0.5444	0.5653	511.8	684.9	1196.7	0.7135	0.7004	1.4139	800.0
820.0	521.03	0.0210	0.5293	0.5503	515.5	680.6	1196.0	0.7171	0.6940	1.4111	820.0
840.0	523.83	0.0211	0.5149	0.5360	519.0	676.4	1195.4	0.7207	0.6877	1.4084	840.0
860.0	526.58	0.0212	0.5013	0.5225	522.6	672.1	1194.7	0.7242	0.6815	1.4057	860.0
880.0	529.29	0.0213	0.4881	0.5094	526.0	667.9	1194.0	0.7277	0.6754	1.4031	880.0
900.0	531.95	0.0213	0.4756	0.4969	529.5	663.8	1193.3	0.7311	0.6694	1.4005	900.0
920.0	534.56	0.0214	0.4635	0.4849	532.9	659.7	1192.6	0.7344	0.6635	1.3980	920.0
940.0	537.13	0.0215	0.4520	0.4735	536.2	655.6	1191.8	0.7377	0.6577	1.3954	940.0
960.0	539.66	0.0216	0.4409	0.4625	539.6	651.5	1191.1	0.7410	0.6520	1.3930	960.0
980.0	542.14	0.0217	0.4303	0.4520	542.8	647.5	1190.3	0.7442	0.6464	1.3905	980.0
1000.0	544.58	0.0217	0.4202	0.4419	546.0	643.5	1189.6	0.7473	0.6408	1.3881	1000.0
1050.0	550.53	0.0219	0.3960	0.4179	554.0	633.6	1187.6	0.7550	0.6273	1.3822	1050.0
1100.0	556.28	0.0222	0.3738	0.3960	561.7	623.9	1185.6	0.7624	0.6141	1.3765	1100.0
1150.0	561.81	0.0224	0.3540	0.3764	569.2	614.3	1183.5	0.7695	0.6014	1.3709	1150.0
1200.0	567.14	0.0226	0.3356	0.3582	576.5	604.9	1181.4	0.7764	0.5891	1.3656	1200.0
1250.0	572.30	0.0228	0.3187	0.3415	583.6	595.6	1179.2	0.7831	0.5772	1.3603	1250.0
1300.0	577.32	0.0230	0.3029	0.3259	590.6	586.3	1177.0	0.7897	0.5654	1.3552	1300.0
1350.0	582.21	0.0232	0.2884	0.3116	597.5	577.2	1174.7	0.7962	0.5540	1.3501	1350.0
1400.0	586.96	0.0235	0.2748	0.2983	604.3	568.1	1172.4	0.8024	0.5428	1.3452	1400.0
1450.0	591.58	0.0237	0.2621	0.2858	611.0	559.1	1170.0	0.8086	0.5318	1.3404	1450.0
1500.0	596.08	0.0239	0.2502	0.2741	617.5	550.2	1167.6	0.8146	0.5212	1.3357	1500.0
1600.0	604.74	0.0244	0.2284	0.2528	630.2	532.6	1162.7	0.8262	0.5003	1.3265	1600.0
1700.0	612.98	0.0249	0.2089	0.2338	642.5	515.0	1157.5	0.8373	0.4801	1.3174	1700.0
1800.0	620.86	0.0254	0.1913	0.2167	654.7	497.2	1151.8	0.8482	0.4601	1.3083	1800.0
1900.0	628.39	0.0260	0.1754	0.2014	666.8	478.9	1145.7	0.8589	0.4402	1.2990	1900.0
2000.0	635.6	0.0265	0.1610	0.1875	679.0	460.0	1139.0	0.8696	0.4200	1.2896	2000.0
2200.0	649.2	0.0277	0.1346	0.1623	703.7	420.0	1123.8	0.8912	0.3788	1.2700	2200.0
2400.0	661.9	0.0292	0.1112	0.1404	729.4	376.4	1105.8	0.9133	0.3356	1.2488	2400.0
2600.0	673.8	0.0310	0.0895	0.1205	756.7	327.8	1084.5	0.9364	0.2892	1.2257	2600.0
2800.0	684.9	0.0333	0.0688	0.1021	786.7	272.3	1058.9	0.9618	0.2379	1.1996	2800.0
3000.0	695.2	0.0367	0.0477	0.0844	823.1	202.5	1025.6	0.9922	0.1754	1.1676	3000.0
3200.0	704.9	0.0459	0.0142	0.0601	887.0	75.9	962.9	1.0461	0.0651	1.1112	3200.0
3226.0	706.1	0.0522	0	0.0522	925.0	0	925.0	1.0785	0	1.0785	3226.0

be higher than that of saturated steam at the same pressure. The relation between pressure and specific volume for dry saturated steam is given by the experimental equation (Goodenough) as:

$$pv^{1.0631} = 484.2 \quad (2)$$

where

p = the pressure in pounds per square inch.
v = the specific volume.

The *total heat* of a dry saturated vapor for any pressure and temperature is the sum of the heat required to raise the temperature of one pound of the liquid from the freezing point to the given temperature and corresponding pressure plus the heat required to entirely vaporize it at this pressure.

Steam which is in contact with the water from which it was generated is called *wet saturated steam* if it contains more or less actual water in the form of mist or priming. The percentage of dry steam in a mixture of steam and water is known as the *quality* (x) of the steam. The total heat of wet vapor at any pressure and temperature is the sum of the heat required to raise the temperature of one pound of the liquid from the freezing point to the given temperature and corresponding pressure plus the heat required to vaporize the part (x) at this pressure.

Chapter 42

HEATING AND VENTILATING STANDARD TERMS

Glossary of Physical and Heating and Ventilating Terms Used in the Text, Standard Abbreviations, Conversion Equations, Drafting Symbols, A.S.H.V.E. Codes

Absolute Humidity: See *Humidity.*

Absolute Temperature: The temperature of a substance measured above *absolute zero.*

Absolute Zero: The temperature (−459.6 F) at which the molecular motion of a substance theoretically ceases. This is the temperature at which the substance theoretically contains no heat energy.

Acceleration: The rate of change of velocity. In the fps system this is expressed in units of one foot per second per second.

$$a = \frac{V}{t}$$

Acceleration Due to Gravity: The rate of gain in velocity of a freely falling body. In the fps system this is 32.174 feet per second per second.

Air Cleaner: A device designed for the purpose of removing air-borne impurities such as dusts, fumes and smokes. (Air cleaners include air washers and air filters).

Air Conditioning: The simultaneous control of all or at least the first three of those factors affecting both the physical and chemical conditions of the atmosphere within any structure. These factors include temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases, and ionization, most of which affect in greater or lesser degree human health or comfort.

Air Infiltration: The inleakage of air through cracks and crevices, and through doors, windows and other openings, caused by wind pressure or temperature difference.

Blast: This word was formerly used to denote forced air circulation, particularly in connection with central fan systems using steam or hot water as the heating medium. As applied in this sense, the word *blast* is now obsolete.

Boiler: A closed vessel in which steam is generated or in which water is heated.

Boiler Heating Surface: That portion of the surface of the heat-transfer apparatus in contact with the fluid being heated on one side and the gas or refractory being cooled on the other, in which the fluid being heated forms part of the circulating system; this surface shall be measured

on the side receiving heat. This includes the boiler, water walls, water screens, and water floor. (*A.S.M.E.* Power Test Codes, Series 1929).

Boiler Horsepower: The equivalent evaporation of 34.5 lb of water per hour from and at 212 F. This is equal to a heat output of $970.2 \times 34.5 = 33{,}471.9$ Btu per hour.

British Thermal Unit: The *mean* British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32 F to 212 F. It is substantially equal to the quantity of heat required to raise 1 lb of water from 63 F to 64 F.

Calorie: The *mean* calorie is $\frac{1}{100}$ of the heat required to raise the temperature of 1 gram of water from *Zero* C to 100 C. It is substantially equal to the quantity of heat required to raise one gram of water from 14.5 C to 15.5 C.

Central Fan System: A mechanical indirect system of heating, ventilating or air conditioning consisting of a central plant where the air is heated and conditioned and then circulated by fans or blowers through a system of distributing ducts.

Chimney Effect: The tendency in a duct or other vertical air passage for air to rise when heated, owing to its decrease in density.

Coefficient of Transmission: The amount of heat (Btu) transmitted *from air to air* in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg Fahrenheit *between the air on the inside and that on the outside of the wall, floor, roof or ceiling.* (The velocity of air passing over the surfaces and the distances from the surfaces at which the temperatures are measured affect the coefficient of transmission).

Column Radiator: A type of direct radiator. (This radiator has not been listed by manufacturers since 1926).

Comfort Line: The effective temperature at which the largest percentage of adults feel comfortable.

Comfort Zone (*Average*)*:* The range of effective temperatures over which the majority (50 per cent or more) of adults feel comfortable. *Comfort Zone* (*Extreme*)*:* The range of effective temperatures over which one or more adults feel comfortable. (See Chapter 2).

Concealed Radiator: See *convector.*

Conductance: The amount of heat (Btu) transmitted from surface to surface in one hour through one square foot of a material or construction, whatever its thickness, when the temperature difference is 1 deg Fahrenheit between the two surfaces.

Conduction: The transmission of heat through and by means of matter unaccompanied by any obvious motion of the matter.

Conductivity: The amount of heat (Btu) transmitted in one hour through one square foot of a homogeneous material *1 in. thick* for a difference in temperature of 1 deg Fahrenheit between the two surfaces of the material.

Conductor (*heat*)*:* A material capable of readily conducting heat. The opposite of an insulator (or insulation).

Convection: The transmission of heat by the circulation of a liquid or a gas such as air. (Convection may be *natural* or *forced*).

Convector: A concealed *radiator*. A heating unit and an enclosure (or shield) located either within, adjacent to, or exterior to the room or space to be heated, but transferring heat to the room or space mainly by the process of convection. (If the heating unit is located exterior to the room or space to be heated, the heat is transferred through one or more ducts or pipes; see Chapter 30).

Decibel: The standard unit for noise or sound intensity. One decibel is the power supply to a telephone receiver which will produce approximately the smallest change in volume of sound which the normal ear can detect.

Degree-Day: A unit representing a difference of 1 deg Fahrenheit existing for one day between the average indoor and outdoor temperatures. The *standard degree-day* is based on an average indoor temperature of 65 F.

Density: The weight of a unit volume, expressed in pounds per cubic foot. ρ (rho) $= \frac{W}{V}$.

Dew-Point Temperature: The temperature corresponding to saturation (100 per cent relative humidity) for a given moisture content.

Diffuser: A vaned device placed at an air supply opening to direct the air flow.

Direct-Indirect Heating Unit: A heating unit located in the room or space to be heated and partially enclosed, the enclosed portion being used to heat air which enters from outside the room.

Direct Radiator: Same as *radiator*.

Direct-Return System (*Hot water*): A hot water system in which the water, after it has passed through a heating unit, is returned to the boiler along a direct path so that the total distance traveled by the water is the shortest feasible, and so that there are considerable differences in the lengths of the several circuits composing the system.

Down-Feed One-Pipe Riser (*Steam*): A pipe which carries steam downward to the heating units and into which the condensation from the heating units drains.

Down-Feed System (*Steam*): A steam heating system in which the supply mains are above the level of the heating units which they serve.

Draft Head (*Side Outlet Enclosure*): The vertical distance of a gravity convector between the bottom of the heating unit and the bottom of the air outlet opening.

Draft Head (*Top Outlet Enclosure*): The vertical distance of a gravity convector between the bottom of the heating unit and the top of the enclosure.

Dry Air: Air with which no water vapor is mixed. This term is used comparatively, since in nature there is always some water vapor included in air, and such water vapor, being a gas, is dry.

Dry-Bulb Temperature: The temperature of the air indicated by

any type of thermometer not affected by the water vapor content or relative humidity of the air.

Dry Return: A return pipe in a steam heating system which carries both water of condensation and air. See *wet return.*

Dust: Solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity, due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

Dynamic Head or Pressure: The total or impact pressure. This is the sum of the radial pressure and the velocity pressure at the point of measurement.

Effective Temperature: An arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air. Effective temperature is a composite index which combines the readings of temperature, humidity, and air motion into a single value. The numerical value of the effective temperature scale has been fixed by the temperature of saturated air which induces an identical sensation of warmth.

Equivalent Evaporation: The amount of water a boiler would evaporate, in pounds per hour, if it received feed water at 212 F and vaporized it at the same temperature and atmospheric pressure.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined. It is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932).

Estimated Maximum Load: Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry. (A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand-Fired Boilers—edition of April 1932).

Extended Heating Surface: See *heating surface.*

Extended Surface Heating Unit: A heating unit having a relatively large amount of extended surface which may be integral with the core containing the heating medium or assembled over such a core, making good thermal contact by pressure or by being soldered to the core or by both pressure and soldering. (An extended surface heating unit is usually placed within an enclosure and therefore functions as a convector).

Fan Furnace System: See *warm air heating system.*

Force: The action on a body which tends to change its relative condition as to rest or motion. $F = \frac{WV}{gt}$.

Fumes: Particles of solid matter resulting from such chemical processes as combustion, explosion, and distillation, ranging from 0.1 to 1.0 micron in size.

Furnace: That part of a boiler or warm air heating plant in which combustion takes place. Also, a fire-pot.

Furnace Volume (*Total*): The total furnace volume for horizontal return-tubular boilers and water-tube boilers is the cubical contents of the furnace between the grate and the first plane of entry into or between tubes. It therefore includes the volume behind the bridge wall as in ordinary horizontal return-tubular boiler settings, unless manifestly ineffective (*i.e.*, no gas flow taking place through it), as in the case of waste-heat boilers with auxiliary coal furnaces, where one part of the furnace is out of action when the other is being used. For Scotch or other internally fired boilers it is the cubical contents of the furnace, flues and combustion chamber, up to the plane of first entry into the tubes. (*A.S.M.E.* Power Test Codes, Series 1929).

Grate Area: The area of the grate surface, measured in square feet, to be used in estimating the rate of burning fuel. This area is construed to mean the area measured in the plane of the top surface of the grate, except that with special furnaces, such as those having magazine feed, or special shapes, the grate area shall be the mean area of the active part of the fuel bed taken perpendicular to the path of the gases through it. For furnaces having a secondary grate, such as those in double-grate down-draft boilers, the effective area shall be taken as the area of the upper grate plus one-eighth of the area of the lower grate, both areas being estimated as defined above. (A.S.H.V.E. Standard and Short Form Heat Balance Codes for Testing Low-Pressure Steam Heating Solid Fuel Boilers).

Gravity Warm Air Heating System: See *warm air heating system.*

Grille: A perforated covering for an air inlet or outlet usually made of wire screen, pressed steel, cast-iron or plaster. (Grilles may be plain or ornamental).

Heat: A form of energy generated by the transformation of some other form of energy, as by combustion, chemical action, or friction. [According to the molecular theory, heat consists of the kinetic and potential energy of the molecules of a substance. The addition of heat energy to a body increases the temperature or the kinetic energy of motion of its molecules (*sensible heat*) or increases their potential energy of position but does not increase the temperature, as when melting or boiling occurs (*latent heat*).]

Heat Capacity: The amount of heat (Btu or calories) required to raise the temperature of a body (of any mass and variety of parts) one degree (Fahrenheit or centigrade). This will depend on the masses and specific heats of the various parts of the body.

Therefore

$$S = m_1 s_1 + m_2 s_2 + m_3 s_3 \ldots \text{ etc.}$$

where

S is the heat capacity and m_1, m_2, m_3, and s_1 s_2, s_3 stand for the masses and corresponding specific heats of the parts, respectively.

Heating Medium: A substance such as water, steam, air, electricity

or furnace gas used to convey heat from the boiler, furnace or other source of heat or energy to the heating unit from which the heat is dissipated.

Heating Surface: The exterior surface of a heating unit. *Extended heating surface (or extended surface):* Heating surface having air on both sides and heated by conduction from the prime surface. *Prime Surface:* Heating surface having the heating medium on one side and air (or extended surface) on the other. (See also *boiler heating surface*).

Horsepower: A unit to indicate the time rate of doing work equal to 550 ft-lb per second or 33,000 ft-lb per minute. (One horsepower = 745.8 watts. In practice this is considered 746 watts).

Hot Water Heating System: A heating system in which water is used as the medium by which heat is carried through pipes from the boiler to the heating units.

Humidity: The water vapor (either saturated or super-heated steam) occupying any space, which may or may not contain other vapors and gases at the same time. *Relative Humidity:* A ratio, although usually expressed in per cent, used to indicate the degree of saturation existing in any given space resulting from the water vapor present in that space. The presence of air or other gases in the same space at the same time has nothing to do with the relative humidity of the space, which depends merely on the temperature and partial pressure of the vapor. *Absolute Humidity:* The actual weight of water vapor contained in a given volume of a mixture of air and water vapor at the observed temperature.

Humidistat: A regulatory device, actuated by changes in humidity, used for the control of humidity.

Hygrostat: Same as humidistat.

Insulation *(heat):* A material having a relatively high heat-resistance per unit of thickness.

Latent Heat: See *heat*.

Mass: The quantity of matter, in pounds, to which the unit of force (one pound) will give an acceleration of one foot per second per second. $m = \frac{W}{g}$.

Mb, Mbh[1]**:** Symbols which represent, respectively, 1000 Btu and 1000 Btu per hour.

Mechanical Equivalent of Heat: The mechanical energy necessary to produce 1 Btu of heat energy. J = 777.5 ft-lb.

Micron: A unit of length, the thousandth part of one millimeter or the millionth of a meter.

Mol: The unit of weight for gases. It is defined as m lb where m denotes the molecular weight of a gas. (For any gas the volume of 1 *mol* at 32 F and standard atmospheric pressure is 358.65 cu ft and the weight of a cubic foot is 0.002788 m lb).

Neutral Zone: The level within a room or building at which the pressure is exactly equal to the outside barometric pressure.

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[1]*These symbols were approved by the* A.S.H.V.E., June, 1933.

One-Pipe Supply Riser (*Steam*)*:* A pipe which carries steam upward to a heating unit and which also carries the condensation from the heating unit in a direction opposite to the steam flow.

One-Pipe System (*Hot water*)*:* A hot water system in which the water flows through more than one heating unit before it returns to the boiler; consequently, the heating units farthest from the boiler are supplied with cooler water than those near the boiler in the same circuit.

One-Pipe System (*Steam*)*:* A steam heating system consisting of a main circuit in which the steam and condensate flow in the same pipe and usually in opposite directions. Ordinarily to each heating unit there is but one connection which must serve as both the supply and the return, although separate supply and return connections may be used.

Overhead System: Any steam or hot water system in which the supply main is above the heating units. (With a steam system the return must be below the heating units; with a water system, the return *may* be above the heating units).

Panel Radiator: A heating unit placed on or flush with a flat wall surface and intended to function essentially as a radiator.

Panel Warming: A method of heating involving the installation of the heating units (pipe coils) within the wall, floor or ceiling of the room, so that the heating process takes place mainly by radiation from the wall, floor or ceiling surfaces to the objects in the room.

Plenum Chamber: An air compartment maintained under pressure and connected to one or more distributing ducts.

Prime Surface: See *heating surface.*

Power: The rate of performing work, expressed in units of horsepower, one of which is equal to 550 ft-lb of work per second, or 33,000 ft-lb per minute.

Psychrometer: An instrument for ascertaining the humidity or hygrometric state of the atmosphere. *Psychrometric:* Pertaining to psychrometry or the state of the atmosphere as to moisture. *Psychrometry:* The branch of physics that treats of the measurement of degree of moisture, especially the moisture mixed with the air.

Pyrometer: An instrument for measuring high temperatures.

Radiation: The transmission of heat through space by wave motion.

Radiator: A heating unit located within the room or space to be heated and exposed to view. (A radiator transfers heat by radiation to objects "it can see" and by conduction to the surrounding air which in turn is circulated by natural convection; a so-called *radiator* is also a *convector* but the single term *radiator* has been established by long usage). *Concealed Radiator:* See convector.

Recessed Radiator: A heating unit set back into a wall recess but not enclosed.

Register: A grille with a built-in damper or shutter.

Relative Humidity: See *humidity:* see also discussion of relative humidity in Chapter 1.

Return Mains: The pipes which return the heating medium from the heating units to the source of heat supply.

Reversed-Return System (*Hot Water*): A hot water heating system in which the water from several heating units is returned along paths arranged so that all circuits composing the system or composing a major subdivision of the system are practically of equal length.

Roof Ventilator: A device placed on the roof of a building to permit egress of air.

Sensible Heat: See *heat.*

Smoke: Carbon or soot particles less than 0.1 micron in size which result from the incomplete combustion of carbonaceous materials such as coal, oil, tar, and tobacco.

Smokeless Arch: An inverted baffle placed in an up-draft furnace toward the rear to aid in mixing the gases of combustion and thereby to reduce the smoke produced.

Specific Gravity: The ratio of the weight of a body to the weight of an equal volume of water at some standard temperature, usually 39.2 F.

Specific Heat: The quantity of heat, expressed in Btu, required to raise the temperature of 1 lb of a substance 1 deg Fahrenheit.

Specific Volume: The volume, expressed in cu-ft, of one pound of a substance. $v = \frac{1}{\rho} = \frac{V}{W}$.

Split System: A system in which the heating and ventilating are accomplished by means of radiators or convectors supplemented by mechanical circulation of air (heated or unheated) from a central point.

Square Foot of Heating Surface (*equivalent*): Equivalent direct *radiation* (EDR). By definition, that amount of heating surface which will give off 240 Btu per hour. (The *equivalent* square foot of heating surface may have no direct relation to the actual surface area).

Stack Height: The vertical distance of a gravity convector between the bottom of the heating unit and the top of the outlet opening.

Standard Air: As defined by A.S.H.V.E. codes, *standard air* is air weighing 0.07488 lb per cubic foot, which is air at 68 F dry-bulb and 50 per cent relative humidity with a barometric pressure of 29.92 in. of mercury. (Most engineering tables and formulae involving the weight of air are based on air weighing 0.07495 lb per cubic foot, which is dry air at 70 F dry-bulb with a barometric pressure of 29.92 in. of mercury. The error involved in disregarding the difference between the above two weights is very slight and in most instances may be neglected).

Static Head or Pressure: The radial pressure within an enclosure, tending to burst it. (This is also termed frictional or resistance pressure or maintained resistance).

Steam: Steam is water vapor which exists in the vaporous condition because sufficient heat has been added to the water to supply the latent heat of evaporation and change the liquid into vapor. Steam in contact with the water from which it has been generated may be *dry-saturated* steam or *wet-saturated* steam. The latter contains more or less actual water in the form of mist. If steam is heated, and the pressure maintained the same as when it was vaporized, its temperature will increase and it will become *superheated.*

Steam Heating System: A heating system in which heat is transferred from the boiler or other source of steam, to the heating units by means of steam at, above, or below atmospheric pressure.

Steam Trap: A device for allowing the passage of condensate and preventing the passage of steam, or for allowing the passage of air as well as condensate.

Superheated Steam: See *steam.*

Supply Mains (*Steam*)*:* The pipes through which the steam flows from the boiler or source of supply to the run-outs and risers leading to the heating units.

Surface Conductance: The amount of heat (Btu) transmitted by radiation, conduction and convection *from a surface to the air or liquid surrounding it*, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg between the surface and the surrounding air or liquid.

Synthetic Air Chart: A chart for evaluating the air conditions maintained in a room.

Thermostat: An instrument which responds to changes in temperature and which directly or indirectly controls the source of heat supply.

Tube (or Tubular) Radiator: A cast-iron heating unit used as a radiator and having small vertical tubes.

Two-Pipe System (*Steam or water*)*:* A heating system in which one pipe is used for the supply of the heating medium to the heating unit and another for the return of the heating medium to the source of heat supply. (The essential feature of a two-pipe system is that each heating unit receives a direct supply of the heating medium which medium cannot have served a preceding heating unit).

Underfeed Distribution System (*Hot water*)*:* A hot water heating system in which the main flow pipe is below the heating units.

Underfeed Stoker: A stoker which feeds the coal underneath the fuel bed.

Unit Air Conditioner: A self-contained air conditioning plant which provides for humidification or dehumidification, air washing, heating or cooling, and includes spray nozzles, air re-heater, fan, pump, automatic control, all within a common enclosure. (As a rule the outlets are designed to accomplish air distribution without ducts).

Unit Cooler: A combination of a cooling coil and a fan or blower, erected as a unit and having a common enclosure. (As a rule the outlets are designed to accomplish air distribution without ducts).

Unit Heater: Any combination of a heating unit and a fan or blower, having a common enclosure and placed within or adjacent to the space to be heated; generally no ducts are attached to the inlets or outlets. (Unit heaters are designed primarily for industrial use).

Unit Ventilator: A unit heater designed to use all or part outdoor air with or without alternate provision for handling recirculated air. (Unit ventilators are intended primarily for school and office ventilation).

Up-Feed System (*Steam*)*:* A steam heating system in which the supply mains are below the level of the heating units which they serve.

Vacuum Heating System: A two-pipe steam heating system equipped with the necessary accessory apparatus which will permit operating the system below atmospheric pressure when desired.

Vapor: Any substance in the gaseous state.

Vapor Heating System: A steam heating system which operates under pressures at or near atmospheric and which returns the condensation to the boiler or receiver by gravity. (Vapor systems have thermostatic traps or other means of resistance on the return ends of the heating units for preventing steam from entering the return mains; they also have a pressure-equalizing and air-eliminating device at the end of the dry return. *Direct Vent Vapor System:* A vapor heating system with air valves which do not permit re-entry of air.

Velocity: The time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of one foot per second. $V = \frac{s}{t}$.

Velocity Head or Pressure: The head or pressure required to create the velocity of flow, that is, the head or pressure required to accelerate the mass from a state of rest to the velocity at the point measured.

Ventilation: The process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned. (See *Air Conditioning*).

Warm Air Heating System: A warm air heating plant consists of a heating unit (fuel-burning furnace) enclosed in a casing, from which the heated air is distributed to the various rooms of the building through ducts. If the motive head producing flow depends on the difference in weight between the heated air leaving the casing and the cooler air entering the bottom of the casing, it is termed a *gravity* system. A booster fan may, however, be used in conjunction with a gravity-designed system. If a fan is used to produce circulation and the system is designed especially for fan circulation, it is termed a *fan furnace* system or a *central fan furnace* system. A fan furnace system may include air washers, and filters.

Wet-Bulb Temperature: The lowest temperature which a water wetted body will attain when exposed to an air current. (This is the temperature of adiabatic saturation).

Wet Return: That part of a return main of a steam heating system which is filled with water of condensation. (The wet return usually is below the level of the water line in the boiler, although not necessarily so).

ABBREVIATIONS[2]

Absolute	abs
Acceleration, due to gravity	*g*
Acceleration, linear	*a*
Air horsepower	air hp
Alternating-current (as adjective)	a-c
Ampere	amp
Ampere-hour	amp-hr

[2]From compilations of abbreviations approved by the *American Standards Association.*

Area.. A
Atmosphere.. atm
Average.. avg
Avoirdupois... avdp
Barometer.. bar.
Boiler pressure... bp
Boiling point.. bp
Brake horsepower... bhp
Brake horsepower-hour... bhp-hr
British thermal unit.. Btu
Calorie... cal
Centigram.. cg
Centimeter... cm
Centimeter-gram-second (system).. cgs
Change in specific volume during vaporization.................................. v_{fg}
Cubic... cu
Cubic foot.. cu ft
Cubic feet per minute... cfm
Cubic feet per second... cfs
Decibel.. db
Degree[3]... deg or °
Degree centigrade... C
Degree Fahrenheit.. F
Degree Kelvin... K
Degree Réaumur... R
Density, Weight per unit volume, Specific weight.............................. d or ρ (rho)

$$\rho = \frac{1}{v}$$

Diameter.. D or diam
Direct-current (as adjective).. d-c
Distance, linear.. s
Dry saturated vapor, Dry saturated gas at saturation pressure and temperature, Vapor in contact with liquid.. *Subscript* g
Entropy (The capital should be used for any weight, and the small letter for unit weight)... S or s
Feet per minute.. fpm
Feet per second.. fps
Foot... ft
Foot-pound... ft-lb
Foot-pound-second (system).. fps
Force, total load... F
Freezing point.. fp
Gallon.. gal
Gallons per minute... gpm
Gallons per second... gps
Gram.. g
Gram-calorie.. g-cal
Head... H or h
Heat content, Total heat, Enthalpy. (The capital should be used for any weight and the small letter for unit weight)... H or h
Heat content of saturated liquid, Total heat of saturated liquid, Enthalpy of saturated liquid, sometimes called heat of the liquid............................. h_f
Heat content of dry saturated vapor, Total heat of dry saturated vapor, Enthalpy of dry saturated vapor.. h_g
Heat of vaporization at constant pressure.. L or h_{fg}
Horsepower.. hp
Horsepower-hour.. hp-hr
Inch... in.

[3]It is recommended that the abbreviation for the temperature scale, F, C, K, etc., be included in expressions for numerical temperatures but, wherever feasible, the abbreviation for *degree* be omitted; as 68 F.

Inch-pound .... in-lb
Indicated horsepower .... ihp
Indicated horsepower-hour .... ihp-hr
Internal energy, Intrinsic energy. (The capital should be used for any weight and the small letter for unit weight) .... U or u
Kilogram .... kg
Kilowatt .... kw
Kilowatthour .... kwhr
Length of path of heat flow, thickness L
Load, total .... W
Mass .... m
Mechanical efficiency .... e_m
Mechanical equivalent of heat .... J
Melting point .... mp
Meter .... m
Micron .... μ (mu)
Miles per hour .... mph
Minute .... min
Molecular weight .... mol. wt
Mol .... mol
Ounce .... oz
Power, Horsepower, Work per unit time .... P
Pressure, Absolute pressure, Gage pressure, Force per unit area .... p
Quantity (total) of fluid, water, gas, heat; Quantity by volume; Total quantity of heat transferred .... Q
Quality of steam, Pounds of dry steam per pound of mixture x
Revolutions per minute .... rpm
Saturated liquid at saturation pressure and temperature, Liquid in contact with vapor .... *Subscript* $_f$
Specific gravity .... sp gr
Specific heat .... sp ht or c
Specific heat at constant pressure .... c_p
Specific heat at constant volume .... c_v
Specific volume, Volume per unit weight, Volume per unit mass v
Square foot .... sq ft
Square inch .... in. sq
Temperature (ordinary) F or C. (*Theta* is used preferably only when t is used for Time in the same discussion) .... t or θ (*theta*)
Temperature (absolute) F abs or K. (Capital *theta* is used preferably only when small *theta* is used for ordinary temperature) .... T or Θ (*capital theta*)
Thermal conductance[4] (heat transferred per unit time, per degree) C

$$C = \frac{1}{R} = \frac{kA}{L} = \frac{q}{t_1 - t_2}$$

Thermal conductance per unit area, unit conductance (heat transferred per unit time per unit area, per degree) .... C_A

$$C_A = \frac{C}{A} = \frac{1}{RA} = \frac{q}{A(t_1 - t_2)} = \frac{k}{L}$$

Thermal conductivity (heat transferred per unit time per unit area, and per degree per unit length) .... k

$$k = \frac{\frac{q}{A}}{\frac{(t_1 - t_2)}{L}}$$

[4]Terms ending *ivity* designate properties independent of size or shape, sometimes called *specific properties*. Examples are—conductivity and resistivity. Terms ending *ance* designate quantities depending not only on the material, but also upon size and shape, sometimes called *total quantities*. Examples are—conductance and transmittance. Terms ending *ion* designate rate of heat transfer. Examples are—conduction and transmission.

Surface coefficient of heat transfer, Film coefficient of heat transfer, Individual coefficient of heat transfer (heat transferred per unit time per unit area, per degree) .... f

$$f = \frac{\frac{q}{A}}{t_1 - t_2}$$

(In general f is not equal to k/L, where L is the actual thickness of the fluid film).

Over-all coefficient of heat transfer, Thermal transmittance per unit area (heat transferred per unit time per unit area, per degree over-all) .... U

$$U = \frac{\frac{q}{A}}{t_1 - t_2}$$

Thermal transmission (heat transferred per unit time) .... q

$$q = \frac{Q}{t}$$

Thermal resistance (degrees, per unit of heat transferred per unit time) .... R

$$R = \frac{t_1 - t_2}{q} = \frac{L}{kA}$$

Thermal resistivity .... $1/k$

Vaporization values at constant pressure, Differences between values for saturated vapor and saturated liquid at the same pressure .... *Subscript* fg

Velocity V

Volume (total) V

Volume per unit time, Rate at which quantity of material passes through a machine, Quantity of heat per unit time, Quantity of heat per unit weight .... q

Watt .... w

Watthour .... whr

Weight of a major item, Total weight .... W

Weight rate, Weight per unit of power, Weight per unit of time .... w

Work (total) .... W

CONVERSION EQUATIONS

Fahrenheit degrees = 9/5 centigrade degrees + 32.

Centigrade degrees = 5/9 (Fahrenheit degrees − 32).

Absolute temperature, expressed in Fahrenheit degrees = Fahrenheit degrees + 459.6. In heating and ventilating work, 460 is usually used.

Absolute temperature, expressed in centigrade degrees = centigrade degrees + 273.1.

Powers and Work

1 ton refrigeration	= 200 Btu per minute
Latent heat of ice	= 143.33 Btu per pound
1 Btu	= 777.5 ft-lb 0.293 watthours 252.02 mean calories
1 watthour	= 2,655.2 ft-lb 3.415 Btu 3600 joules 860.648 mean calories
1 mean calorie	= 0.003968 Btu 3.085 ft-lb 0.0011619 watthours

1 kilowatt (1000 watts)	= 1.3405 horsepower 56.92 Btu per minute 44,252.7 ft-lb per minute
1 horsepower	= 0.746 kilowatt 42.44 Btu per minute 33,000 ft-lb per minute 550 ft-lb per second
1 boiler horsepower	= 33,471.9 Btu per hour

Weight and Volume

1 gal (U. S.)	= 231 cu in. 0.13368 cu ft
1 British or Imperial gallon	= 277.274 cu in.
1 cu ft	= 7.4805 gal 1728 cu in.
1 cu ft water at 60 F	= 62.37 lb
1 cu ft water at 212 F	= 59.76 lb
1 gal water at 60 F	= 8.34 lb
1 gal water at 212 F	= 7.99 lb
1 lb (avdp)	= 16 oz 7000 grains
1 bushel	= 1.244 cu ft
1 short ton	= 2000 lb
1 long ton	= 2240 lb

Pressure

1 lb per square inch	= 144 lb per square foot 2.0416 in. mercury at 62 F 2.309 ft water at 62 F 27.71 in. water at 62 F
1 oz per square inch	= 0.1276 in. mercury at 62 F 1.732 in. water at 62 F
1 atmosphere	= 14.7 lb per square inch 2116.3 lb per square foot 33.974 ft water at 62 F 30 in. mercury at 62 F 29.921 in. mercury at 32 F
1 in. water at 62 F	= 0.03609 lb per square inch 0.5774 oz per square inch 5.196 lb per square foot
1 ft water at 62 F	= 0.433 lb per square inch 62.355 lb per square foot
1 in. mercury at 62 F	= 0.491 lb per square inch 7.86 oz per square inch 1.131 ft water at 62 F 13.57 in. water at 62 F

Metric Units

1 cm	= 0.3937 in.
1 in.	= 2.54 cm
1 m	= 3.281 ft
1 ft	= 0.3048 m
1 sq cm	= 0.155 sq in.
1 sq in.	= 6.45 sq cm
1 sq m	= 10.765 sq ft
1 sq ft	= 0.0929 sq m
1 cu cm	= 0.061 cu in.

1 cu in.	= 16.39 cu cm
1 cu m	= 35.32 cu ft
1 cu ft	= 0.0283 cu m
1 liter	= 1000 cu cm = 0.264 gal
1 kg	= 2.2046 lb
1 lb	= 0.4536 kg
1 metric ton	= 2205 lb (avdp)
1 gram	= 980.59 dynes = 0.002205 lb
1 kilometer per hour	= 0.6214 mph
1 gram per square centimeter	= 0.0290 in. mercury, at 0 deg C 0.394 in. water, at 15 C
1 kg per square centimeter (metric atmosphere)	= 14.22 lb per square inch
1 gram per cubic centimeter	= 0.03614 lb per cubic inch 62.43 lb per cubic foot
1 dyne	= 0.00007233 poundals
1 joule	= 10,000,000 ergs 0.73767 ft-lb
1 metric horsepower	= 75 kg-m per second 0.986 hp (U. S.)
1 kilogram-calorie (large calorie)	= 1000 gram-calories (small calorie) 3.97 Btu
1 kilogram-calorie per kilogram	= 1.8 Btu per pound
1 gram-calorie per square centimeter	= 3.687 Btu per square foot
1 gram-calorie per square centimeter per centimeter	= 1.451 Btu per square foot per inch
1 gram-calorie per second per square centimeter for a temperature graduation of 1 deg C per centimeter	= 2903 Btu per hour per square foot for a temperature graduation of 1 deg F per inch of thickness.

SYMBOLS FOR HEATING AND VENTILATING DRAWINGS[5]

1. The objects of this standard set of symbols are to insure the correct interpretation of drawings and to conserve drafting room time by establishing simple and unmistakable symbols for the component parts of the heating and ventilating systems. In preparing the list of symbols an effort has been made to follow existing practice in so far as possible but the list cannot be expected to match exactly the existing practice of every drafting room.

2. Simplicity, ease of execution and unmistakable identification were carefully considered in selecting the symbols. Uncommon fittings and appliances such as vacuum pumps, separators, etc., have purposely been omitted in order to produce a list which can be easily remembered. It is assumed that when the scale of the drawing permits, the valves and fittings will be drawn to scale and a conventional representation is then unnecessary.

3. High pressure steam supply pipe

4. Low pressure steam supply pipe

5. Hot water pipe—flow

6. Return pipe—steam or water

7. Air vent line

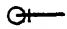
[5]From A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings, edition of 1929.

8. Flanges 

9. Screwed union 

10. Elbow 

11. Elbow—looking up 

12. Elbow—looking down 

13. Tee 

14. Tee—looking up 

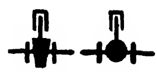
15. Tee—looking down 

16. Gate valve 

17. Globe valve 

18. Angle valve 

19. Angle valve—stem perpendicular 

20. Lock shield valve 

21. Check valve 

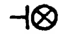
22. Reducing valve 

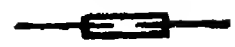
23. Diaphragm valve 

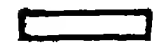
24. Diaphragm valve—stem perpendicular 

25. Thermostat 

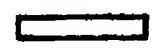
26. Radiator trap—elevation 

27. Radiator trap—plan 

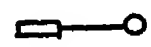
28. Expansion joint 

29. Column radiator—plan 

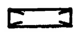
30. Column radiator—elevation 

31. Wall radiator—plan 

32. Wall radiator—elevation 

33. Pipe coil—plan 

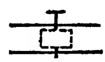
34. Pipe coil—elevation 

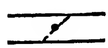
35. Indirect radiator—plan 

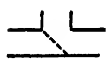
36. Indirect radiator—elevation 

37. Supply duct—section 

38. Exhaust duct—section 

39. Butterfly damper—plant (or elevation) 

40. Butterfly damper—elevation (or plan) 

41. Deflecting damper—square pipe 

42. Vanes 

43. Air supply outlet 

44. Exhaust outlet 

A.S.H.V.E. CODES

Various codes and standards relating to the design, installation, testing, rating and maintenance of materials and equipment used for the heating and ventilation of buildings, have been adopted by the Society as follows:

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Air purity	Synthetic Air Chart	June, 1917	A.S.H.V.E. TRANSACTIONS, Vol. 23, p. 607, and THE GUIDE, 1931
Boilers (testing)	Standard and Short-Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers (Codes 1 and 2)	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929.
Boilers (testing)	A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers (Code 3)[a]	June, 1929	A.S.H.V.E. TRANSACTIONS, Vol. 35, 1929.
Boilers—Oil Fuel (testing)	A.S.H.V.E. Standard Code for Testing Steam Heating Boilers Burning Oil Fuel	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931
Boilers (rating)	A.S.H.V.E. Standard Code for Rating Steam Heating Solid Fuel Hand Fired Boilers	January, 1929 Revised April, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 42
Convectors	A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code)	January, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 37, 1931, p. 367
Ethics	Code of Ethics for Engineers	January, 1922	A.S.H.V.E. TRANSACTIONS, Vol. 28, 1922, p. 6 (See frontispiece THE GUIDE, 1934)
Fans	Standard Test Code for Disc and Propeller Fans, Centrifugal Fans and Blowers	May, 1923. Revised June, 1931	A.S.H.V.E. TRANSACTIONS, Vol. 29, 1923, p. 407[b]
Garages	Code for Heating and Ventilating Garages	June, 1929	A.S.H.VE. TRANSACTIONS, Vol. 35, 1929, p. 355
Heat transmission through walls	Standard Test Code for Heat Transmission through Walls	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 34, 1928, p. 253
Minimum requirements	Code of Minimum Requirements for Heating and Ventilation of Buildings, Edition–1929	June, 1925	A.S.H.V.E. Codes

[a]Originally adopted by the *National Boiler and Radiator Manufacturers Association.*
[b]Also, see *Heating, Piping and Air Conditioning,* August, 1931, p. 743.

SUBJECT	TITLE	WHEN ADOPTED	REFERENCE
Pitot tube	Code for Use of Pitot Tube	January, 1914	A.S.H.V.E. TRANSACTIONS, Vol. 20, 1914, p. 211
Radiators	Code for Testing Radiators	January, 1927	A.S.H.V.E. TRANSACTIONS, Vol. 33, 1927, p. 18
Unit heaters	Standard Code for Testing and Rating Steam Unit Heaters[c]	January, 1930	A.S.H.V.E. TRANSACTIONS, Vol. 36, 1930, p. 165
Unit Ventilators	A.S.H.V.E. Standard Code for Testing and Rating Steam Unit Ventilators	June, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 25
Ventilation	Report of Committee on Ventilation Standards	August, 1932	A.S.H.V.E. TRANSACTIONS, Vol. 38, 1932, p. 383

The following Codes and Standards have been endorsed or approved by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS:

SUBJECT	TITLE	SPONSORED BY	REFERENCE
Chimneys	Standard Ordinance for Chimney Construction	*National Board of Fire Underwriters*	Chapter 14, THE GUIDE, 1931
Piping systems	Identification of Piping Systems[d]	*American Society of Mechanical Engineers*	*Heating, Piping and Air Conditioning,* July, 1929
Warm air furnaces	Standard Code Regulating the Installation of Gravity Warm Air Furnaces in Residences	*National Warm Air Heating Association*	*National Warm Air Heating Association,* Columbus, Ohio

[c]Adopted jointly by the *Industrial Unit Heater Association*, and the A.S.H.V.E.

[d]Adopted November, 1928, Sponsored by (1) *American Society of Mechanical Engineers*, (2) *National Safety Council.*

INDEX

TECHNICAL DATA SECTION

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CROSS REFERENCE TO SUBJECTS IN CHAPTERS 1-42 ALPHABETICALLY LISTED

Catalog Data Section

(Pages 615–779)

with an

INDEX TO MODERN EQUIPMENT

(Pages 780–787)

and

INDEX TO ADVERTISERS

(Pages 788–790)

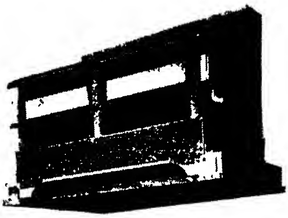
Air-Way Electric Appliance Corporation

Air Conditioning Division

GENERAL OFFICES AND FACTORY

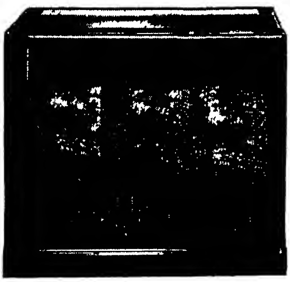
2101 Auburn Avenue, **Toledo, Ohio**

COMPLETE AIR CONDITIONING SYSTEMS (Unit Type)
DOMESTIC AND COMMERCIAL HEATING SYSTEMS (Steam and Hot Water)
INDUSTRIAL UNIT HEATERS (Steam and Hot Water)



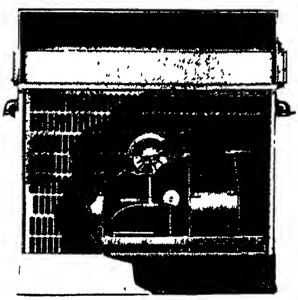
AERIET AIR CONDITIONER—(Ceiling Type)

A compact self-contained overhead unit for stores, offices, restaurants, fur vaults, etc., with limited floor space. Combines every facility for complete air conditioning. Filters, washers, heats, cools, humidifies or dehumidifies. Manual or automatic control. Economical to install and operate. An ideal all year air conditioner for commercial use.



AERIET AIR CONDITIONER—(Floor Type)

Designed for all year air conditioning in homes, offices, etc. Encased in attractive beautifully finished furniture steel cabinet containing complete air conditioning facilities, when connected to steam, refrigeration and electric lines. Cools and dehumidifies in summer, heats and humidifies in winter. Filters, washes and circulates conditioned air throughout the year. Units may be operated singly or in multiple according to space and atmospheric requirements.

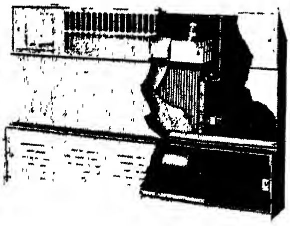


CHASSIS OF AERIET AIR CONDITIONER

Constructed throughout of welded aluminum sheets. Light, durable, strong, non-corrosive, Monel, bronze, stainless steel or copper are used where advisable. Spun glass filters in aluminum frames act as dry air filters and moisture eliminators. Motor fan and pump assembly is compact, strong, silent in operation. All moving parts are readily accessible and interchangeable. Large centrifugal atomizers cannot clog or get out of order.

AERIET HEATING UNITS—
(Steam and Hot Water)

A complete line of mechanical convection units for operation on steam and hot water lines, providing accurate, flexible control of heat distribution. Furnished in cabinets for concealed installation, also in attractive free standing cabinets for radiator replacements. Furnished in a wide range of capacities and sizes.



AIRWAY-UNIVERSAL REFRIGERATING UNITS

Capacities from one to ten tons. Designed especially for use with Aeriet Air Conditioners. Compact, completely self-contained assemblies of compressor with full automatic controls, water cooling coils in insulated tank, cold water circulating pump, automatic condensate elimination.

Write for descriptive folders and engineering data.

American Blower Corporation

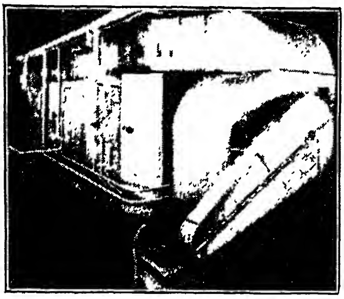
General Offices: **Detroit**

Works: DETROIT, MICHIGAN

Canadian Sirocco Company, WINDSOR, ONTARIO

Branch Offices in Principal Cities of United States and Canada

American Blower Air Conditioning Equipment for the comfort of human beings and efficiency of production in industry



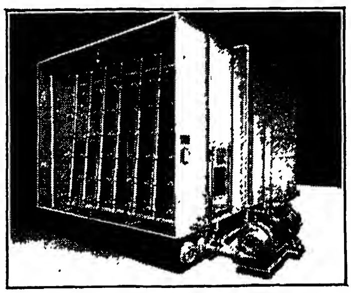
A typical Sirocco System of Air Conditioning for large buildings, auditoriums or factories

Sirocco Systems

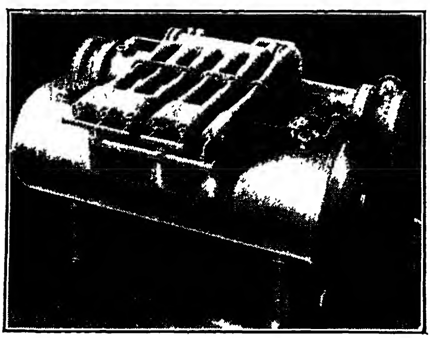
Sirocco Systems for cooling, heating, humidifying, dehumidifying and purifying air, in all classes of business and various manufacturing processes, permits control, as desired, of temperature, humidity and air motion. Component parts: Sirocco or American HS fan and motor, dehumidifier, tempering coils, preheaters, reheaters, spray and circulating water pumps, temperature and humidity control apparatus and a "Ross" Decalorator, or mechanical refrigeration with accessories. Bulletin No. 2727.

Sirocco Dehumidifiers

This line of dehumidifiers, air washers, scrubbers and purifiers, commonly referred to as the "Sirocco Washer," is available in standard sizes and designs, constructed of galvanized iron casing and eliminators with welded iron tank. Capacity: 1,000 to 300,000 cu. ft. of air per minute. These washers are built in lengths from 4′ 0″ to 24′ 0″ with one to six stages of sprays depending upon the duty to be performed. Also built of special metals when required. Bulletin No. 3523.



Sirocco Dehumidifiers and Washers



The "Decalorator" for cooling water in connection with Sirocco Air Conditioning Systems

Decalorators

Decalorators are built and furnished in sizes varying in cooling capacity from 24,000 B.t.u. to 48,000,000 B.t.u. per hour. Chilled or decalorized water is produced by the practical application of a well known physical law, namely: water under high vacuum will vaporize at low temperatures. Decalorators will produce chilled water at a temperature of 38° F. or above. The Decalorator has no moving parts and water is the only refrigerant used. Completely described in Bulletin No. 2927.

American Blower Corporation

General Offices: **Detroit**

Works: DETROIT, MICHIGAN

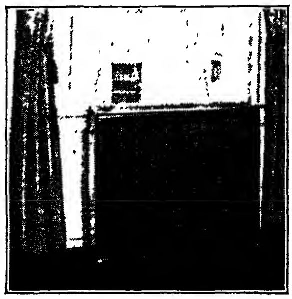
Canadian Sirocco Company, WINDSOR, ONTARIO

Branch Offices in Principal Cities of United States and Canada

Manufacturers of Air Conditioning, Heating, Ventilating, Drying, Dust Collecting, Dust Separation, Mechanical Draft, Pneumatic Conveying and all types of air handling equipment for more than 50 years.

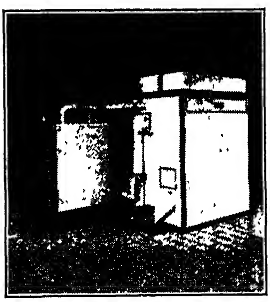
Sirocco Conditioners Series "O"

Series "O" Sirocco Conditioners are made in four sizes for comfort cooling of restaurants, shops, hospitals, offices, etc. Units are provided with three-speed switch automatically controlled by means of a thermostat, and can be equipped with wall box for outside air intake. Air filter and humidifier furnished extra. For winter heating, units are connected to hot water boiler. Complete data in Bulletin No. 2127.



Series "O" Conditioner for cooling and heating offices, restaurants and hospitals, etc.

Sirocco Conditioner Series "R"

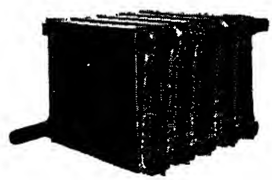


Series "R" Conditioner for Homes

The Series "R" Conditioner is a complete conditioning unit for controlling the temperature, humidity, motion and cleanliness of air in the home. For winter heating, is directly connected to hot water boiler (gas, oil or coal fire). For summer cooling, uses tap, ice cooled or refrigerated water. Equipped with a Sirocco electrical control for regulation of temperature and humidity. Conditioned air is distributed by means of ducts. For details refer to Bulletin No. 1127.

Sirocco Surface Coolers Series "C"

Sirocco Surface Coolers can be easily and quickly installed to add comfort cooling to many types of ventilating systems which are already installed and in operation. A complete Sirocco Surface Cooler consists of casing, tank, eliminators and coils. Detailed information and application data is contained in Bulletin No. 3423.

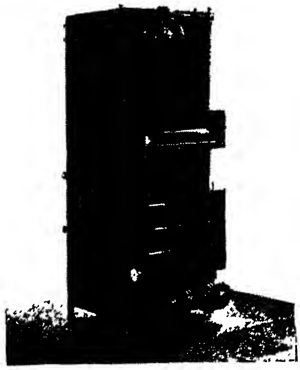


Surface Coolers for low cost comfort cooling

Carrier

Industrial Weathermaker Unit

A complete line of self-contained units for industrial and comfort conditioning, for humidifying or de-humidifying. Each unit is a compact system in itself requiring only heating or refrigeration connections. Equipped for surface or spray type cooling. Capacities range from 1,000 to 10,000 c.f.m. Available in horizontal, vertical, or suspended models. Automatic or manual control. Flexible, easily installed. Perform all the essential functions of a complete central station system.

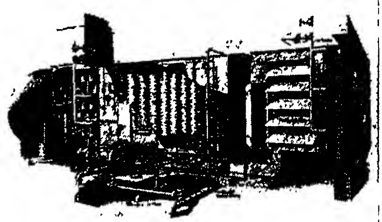


Vertical Type Air Conditioning Unit for control of weather conditions in manufacturing plants. Humidifies, de-humidifies, or heats: washes and circulates the air.

Carrier

Central Plant Weathermaker

A specially designed standardized assembly to meet the increasingly wide variety of applications in central station air conditioning systems. These units are offered as a part of a complete air conditioning system, with automatic dew-point control, pre-heater and re-heater sections, spray chamber, damper control fans and air ducts. Standard for all Carrier air conditioning systems in theatres, office buildings, large stores and public buildings. Installed under licenses of Auditorium Conditioning Corporation.



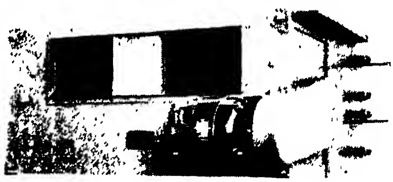
Complete de-humidifying unit of the kind used in central station air conditioning work, showing air intakes, heating coils, spray heads, fan and control devices.

Carrier

Store Weathermaker

(Type 39F)

Especially designed as a suspended unit for stores of all kinds. Air filtering, humidifying and de-humidifying, heating and cooling positive air distribution and automatic control—all built into single attractively encased unit, requiring only connections to source of heat or refrigeration. A thoroughly practical all-year-round system for retail stores, restaurants, small auditoriums, etc.

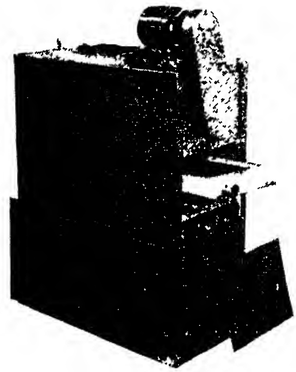


View of Store Air Conditioning Unit. Suspended type. Contains all mechanism for cooling and de-humidifying, circulating and filtering the air.

Carrier

Store Weathermaker
(Type 39E)

Floor mounted model especially designed for stores of all kinds. Units available in models ranging from 800 to 4800 c.f.m. Air filtering, humidifying and de-humidifying, heating and cooling positive air distribution and automatic control—all built into single attractively encased unit, requiring only connections to source of heat or refrigeration. A thoroughly practical all-year-round system for retail stores, restaurants, small auditoriums, etc.

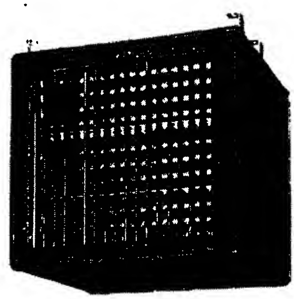


View of Store Air Conditioning Unit complete with casing and ready for refrigerating connections, chamber, heater coils, air filter, cooling coils, re-heater and fan.

Carrier

Store Weathermaker
(Type 39D)

For offices and small stores this suspended type comfort cooling unit supplements the larger Carrier store cooling systems. It is made in one standard size—1½ tons refrigerating effect—and its overall dimensions are 22″x22″x20″ making it flexible, compact and easily installed. It is cleverly designed to harmonize with any style of interior decoration. The fan is of the disc type entirely enclosed in the housing.



Small cooling unit or comfort conditioning unit for offices, stores and shops.

Carrier

Room Weathermaker
(Cabinet Type)

A combination heating and cooling unit for individual rooms in homes or offices. Encased in attractive jacket, it is handsome in appearance, harmonizes with room surroundings. Heats and humidifies in winter, cools and de-humidifies in summer. Replaces a radiator in connection with the heating system and connects to source of refrigeration for summer cooling. Contains air filter, humidifying sprays, fan, heating and cooling coils. One or more cabinets may be installed to meet requirements.

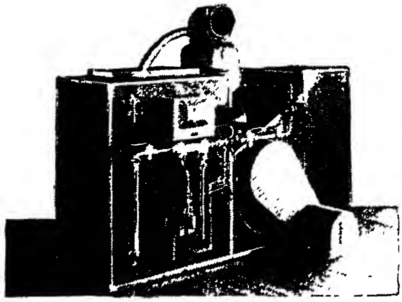
Cabinet model combination cooling and heating unit for rooms and offices, equipped with fans, air filter, spray humidifier, heating and cooling coils.

Carrier

Home Weathermaker

(Direct Fired Type)

The original complete air conditioning system for homes in winter. A gas-fired centrally operated unit, fully automatic, with electric temperature and humidity control. Positive distribution of clean conditioned air. Available in five sizes from 100,000 to 300,000 B.T.U. output per hour. Manufactured or natural gas. Approved by American Gas Association Testing Laboratories.



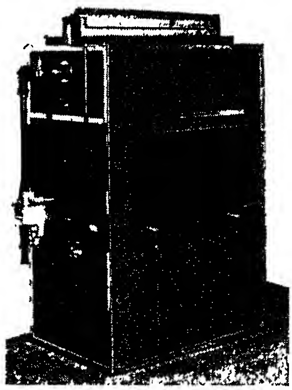
Automatic gas-fired air conditioning plant for homes. Provides controlled humidity and forced circulation of cleaned air with aoutomatic temperature control.

Carrier

Home Weathermaker

(Indirect Fired Type)

An air conditioning plant for homes, employing steam generated in separate boiler with gas, oil or coal as the heating medium. Contains heating coil, filters, humidifier, fan and automatic controls. Connects to supply and return air ducts. Available in capacities from 100,000 to 300,000 B.T.U. per hour output and in five different sizes. Where it is desired to use both air conditioning ducts and direct radiation in the same installation, the Indirect Fired Home Weathermaker is particularly applicable.

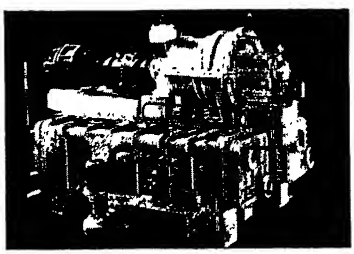


This air conditioning unit is supplied with steam from a separate source and performs the same functions as the machine illustrated at the top of the page.

Carrier

Centrifugal Refrigerating Machine

The most outstandingly successful and highly efficient type of large refrigerating machine for central station air conditioning work—comfort or industrial applications. Available in capacities from 25 to 350 tons. Steam turbine or motor driven. Adapts itself automatically to the air conditioning load. Occupies two-thirds less space for the same capacity than any other system. Operates below atmospheric pressures with Carrene, the harmless, odorless refrigerant assuring absolute safety.

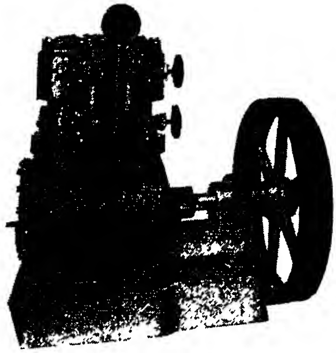


Self-contained refrigeration system including compressor, evaporator and condenser. The most efficient type of refrigerating machine yet developed.

Carrier

Industrial and Marine Refrigeration

Carrier makes a full line of refrigerating machines for every industrial, commercial and marine requirement, using Ammonia, Carbon Dioxide, Freon, Methyl Chloride, Carrene and special refrigerants. Capacities range from $\frac{1}{20}$th to 120 tons A.S.R.E. rating. For air conditioning service, capacities range from $\frac{1}{10}$th to 350 tons. Whatever your refrigerating problem, whether in a small meat market, a great industrial plant or a merchant vessel, you will find a Carrier machine available for your purpose. The Carrier line is the most complete that the industry offers and is built on over 30 years of practical refrigeration experience.

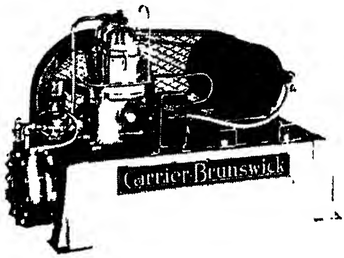


Belt driven Ammonia compressor of from 12 to 24 tons capacity. This unit is extremely compact and represents the latest refinements in refrigeration machine design.

Carrier

Commercial Refrigeration

The Carrier-Brunswick Refrigerating Unit here illustrated, represents one type in a complete line of small machines available in sizes from $\frac{1}{4}$ to 3 h.p. These units are applicable to comfort air conditioning when connected to Carrier atmospheric cabinets and small store units and for commercial refrigeration purposes in meat markets, delicatessen, groceries, restaurants, for dairy products, water cooling, etc.



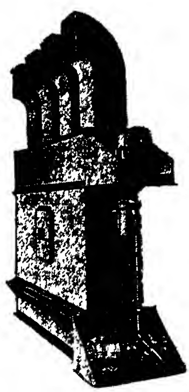
Small refrigerating unit for use with room cooling equipment and for general commercial refrigerating applications.

Carrier

Cold Diffuser

(Floor Mounted Type)

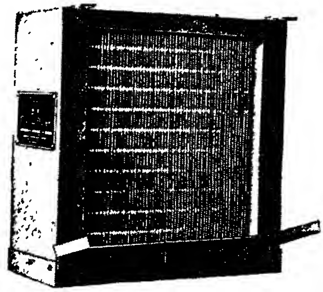
Employs the principle of ductlessly distributed air, positively circulated and automatically controlled as to temperature and humidity. These cold diffusion units which are available in sizes from 1200 to 11,000 c.f.m. and capacities from $\frac{1}{3}$ to 15 tons, are being used in many commercial air conditioning applications—meat packing, fur storage, dairy products, etc. Surface cooling or brine spray units are available.



This cold diffuser finds wide application in meat storage plants, for dairy products cooling and meets many other commercial cooling problems.

Carrier
Cold Diffuser
(Suspended Type)

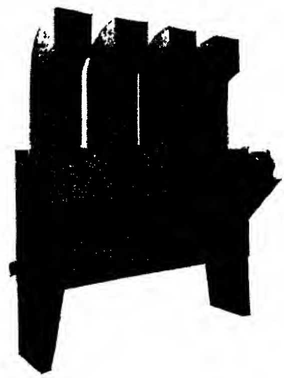
Similar in principle to the floor mounted type of cold diffuser previously described but made in smaller capacities from ¼ to 5 tons. Compact and easy to handle, they take up little space and assure efficient results. Made for single, double and triple fan assemblies—220 to 3800 c.f.m.



Where space saving is unusually important, this suspended type cold diffuser meets a long-felt want.

Carrier
Heat Diffusing Unit
(Floor Mounted Type)

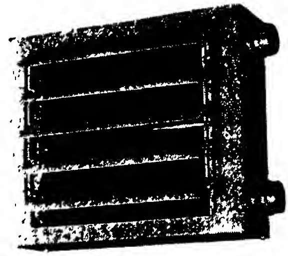
The Carrier line of floor mounted type Industrial Heating Units (formerly known as the York Super-Control Heat Diffusing Unit) embody the most advanced design in unit heater construction and come in a complete range of sizes from 1600 to 16,000 c.f.m. and in two, three and four fan assemblies. Rated from approximately 100,000 to 1,000,000 B.T.U. per hour.



The floor mounted model Heat Diffusing Unit shown above is the most advanced type of equipment for industrial use.

Carrier
Heat Diffusing Unit
(Suspended Type)

A convenient, flexible system for all kinds of factory and industrial heating, will be found in the Carrier Heat Diffusing Unit, "Kroy" type, here illustrated. These units which range from 400 to 5,000 c.f.m. are of the suspended type and take up little space. They are available in one, two or three fan assemblies. Designed for low outlet temperatures, they concentrate maximum heat in the working zone. Rated from 20,000 to 574,000 B.T.U. per hour.



Unusually efficient type of industrial heating unit for controlled zone heating in manufacturing plants.

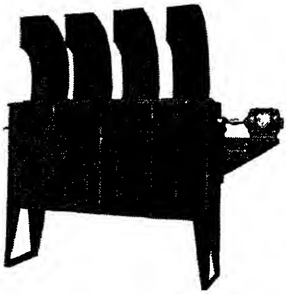
Clarage Fan Company

MAIN OFFICE AND PLANTS

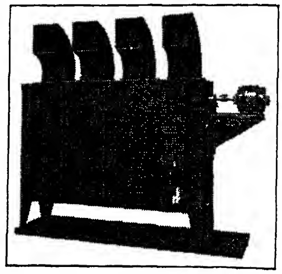
Kalamazoo, Michigan

Sales Engineering Offices in All Principal Cities (Consult Telephone Directory)

Unit Conditioner



Unitherm Unit Heater



Unitherm Unit Cooler

AIR HANDLING AND CONDITIONING EQUIPMENT

Unitherm Unit Heaters—either floor mounted or suspended units. Standard equipment includes Synchrotherm Control (protected by patent coverage) which gives effective heating with low temperature air and marked savings in fuel cost.

Unitherm Unit Coolers—for product or space cooling and refrigeration, producing practically any desired temperature. Units easily installed, saving labor and materials, eliminating expensive bunkers and wall coils.

Unit Conditioners—for cooling and dehumidifying, heating and humidifying, maintaining any desired condition. Offer great flexibility, close control, eliminate duct system—save in both first and operating costs.

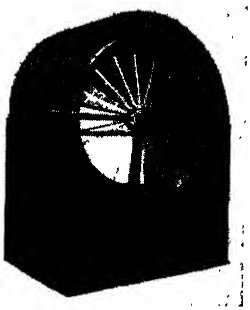
Air Washers—for air purifying, cooling, etc. Improved spray nozzles make possible very economical operation. Various types and sizes to meet all requirements.

Vortex Control—outstanding improvement, with its applications to systems covered by patents granted and pending. Gives any desirable capacity regulation, with the fan operating at constant speed. Does not waste power. Eliminates need for expensive variable speed motor. Adaptable to any Clarage fan for practically any service.

Fans and Blowers—complete line for air conditioning, ventilating, mechanical draft, pneumatic conveying, etc.

Write for Clarage Catalogs.

Air Washer



Vortex Control—lower operating cost than damper control; more efficient; no power wastage. Lower first cost than variable speed motor; simple; inexpensive; trouble-free. Installed in fan inlet as shown above. Automatically controlled, or manually operated.

Boston
New York
Philadelphia
Baltimore
Washington
Charlotte
Atlanta
Palatka
New Orleans
Dallas
Los Angeles

Frick Company

(Incorporated)

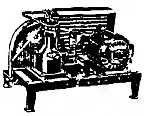
Refrigerating and Ice-Making Machinery

Waynesboro, Penna.

Distributors in 85 FRICK **Principal Cities**

Seattle
Oklahoma City
Memphis
St. Louis
Kansas City
Chicago
Detroit
Cleveland
Cincinnati
Pittsburgh
Buffalo

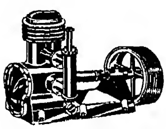
Low pressure Refrigeration



Low Pressure Refrigerating Unit

Commercial Units in more than 20 sizes, with motors of ¼ hp. and up. Charged with either methyl chloride or freon. Air and water cooled condensers. Finned coils, fan units, ice cube makers, beverage coolers, farm milk cabinets, etc. to suit any need. Bulletin No. 97-A.

Freon Refrigeration



Frick Enclosed Freon Compressor

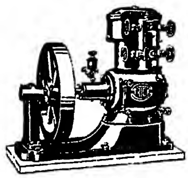
Special freon compressors: vertical enclosed machines, in sizes to suit all air conditioning jobs. Pressure lubricated from pump inside crankcase. Ample gas ports and valves: large capacity, compact design, perfection of details. Coils, coolers, condensers and controls for systems using this gas.

Condensers of All Types

Multipass Shell and Tube Condenser

Either vertical or horizontal shell - and - tube designs, double-pipe or atmospheric, or fan type, for ammonia, carbon dioxide, methyl chloride or freon. Separators, receivers, etc. Bulletins 228, 230 and 232.

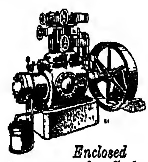
Ammonia Refrigeration



Enclosed Ammonia Compressor

Machines in capacities from ½ ton refrigeration up. Combined units, vertical enclosed type compressors, horizontal machines: for any kind of drive. Full automatic, semi-automatic, or hand control. Widely used in air conditioning. Ask for Bulletins 102, 104, 108, 112 and 138.

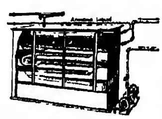
Carbon Dioxide Refrigeration



Enclosed Compressor for Carbon Dioxide

Compressors of the vertical enclosed design, with extra long pistons extending into well lubricated guides in the crankcase: wrist pins are engaged on both sides by forked connecting rods. Smooth-running, efficient and reliable machines, in use the country over. 6 sizes, any drive. Bulletins. Nos. 118, 124 and 208.

Coils and Coolers



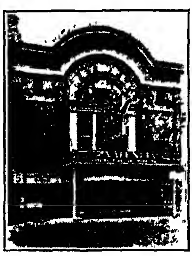
Instant Water Cooler

Continuous welded coils, with or without fins, furnished to suit any refrigerant and any requirement.

Shell - and - tube, vertiflow, and zig-zag "Instant" coolers for water, brine, air, etc. Also complete line of valves and fittings.

Typical Air Conditioning Installations using Frick Refrigeration

Wil-Low Cafeteria, New York City



Valencia Theatre, Chicago

Ed. Stern Printing Plant, Philadelphia

Frigidaire Sales Corporation

Subsidiary of General Motors Corporation

Dayton, Ohio

Outlets in All Principal Cities

Air Conditioning Division

Frigidaire refrigerating equipment has been applied to air conditioning since 1925. In the ensuing years, thousands of Frigidaire installations have been made, covering practically every type of air conditioning application. Along with this long and varied experience, supplemented by both Frigidaire and General Motors research, a comprehensive line of Frigidaire air conditioning apparatus has been evolved.

With equipment to meet the diverse situations arising in the air conditioning field, the Frigidaire organization is prepared and qualified to handle all kinds of air conditioning requirements in residences, stores, offices, hotels, hospitals, railway vehicles, yachts, factories, etc.

To the problems of air conditioning, Frigidaire is devoting the same inventive and engineering genius which has achieved leadership in domestic and commercial refrigeration. This is evidenced by the perfection of Frigidaire air conditioning equipment and engineering; also by the development of FREON, the ideal refrigerant, specifically for air conditioning purposes.

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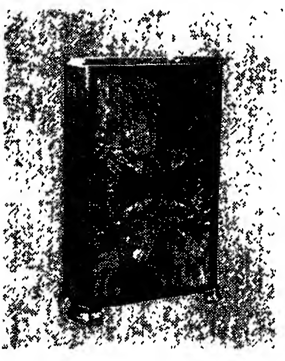
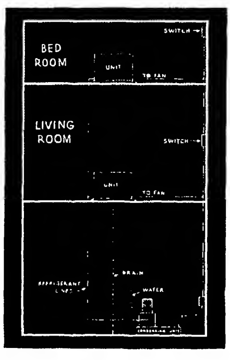
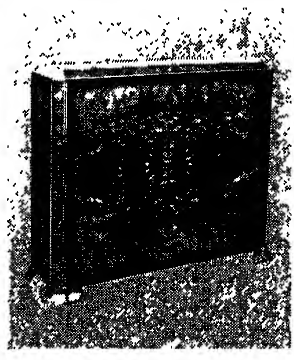


Frigidaire Air Conditioning Cabinet, Vertical Design

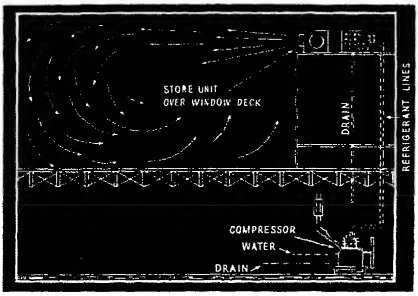


Sketch Showing Two Cabinets Installed for Selective Operation

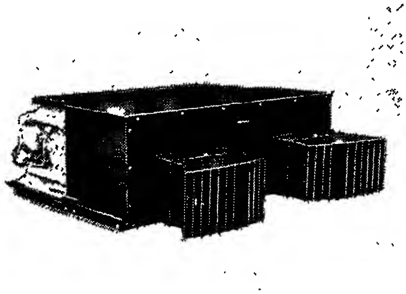


Frigidaire Air Conditioning Cabinet, Horizontal Design

Frigidaire Cabinet Type Air Conditioners. These are of two classes. One class consists of completely self-contained units for cooling, dehumidifying, cleansing, and circulating the air of a room. The other class consists of units for connection to remote compressors. They may be operated individually or in multiples. In the latter event, any selected number or group of the conditioners may be refrigerated at one time. The cabinets of the second class are attachable to steam or hot water lines, in which event the air of the room is cooled in summer and warmed in winter, as well as properly dehumidified or humidified, cleansed, and circulated. All of these may be installed to use any desired percentage of outside air.



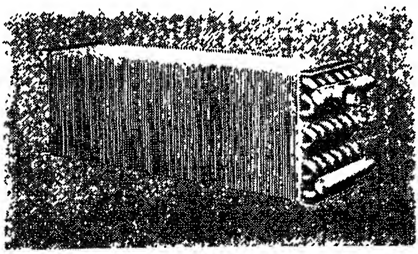
Sketch Showing Model SU-3 Installed Over Window Deck of Store



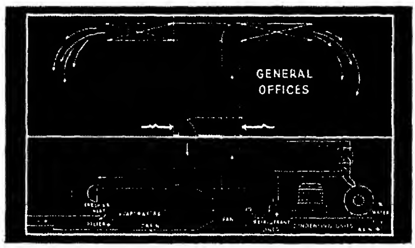
Frigidaire Store Type Air Conditioner, Model SU-3

Frigidaire Store Type Air Conditioners. These comprise cooling coils (or both cooling and heating coils) and fans in housings which have outlets scientifically designed for wide-spread air diffusion. Cooling capacity ranges up to 3 tons of refrigeration per 24 hours. Conditioners of this type may be installed on ceilings, on window decks, on platforms, in alcoves, in the bulkheads of railway cars, etc. Refrigeration is supplied by remotely installed Frigidaire compressors. Several units may be connected to a single compressor. Provisions are made for drawing in as large a proportion of outside air as desirable.

• • •



Frigidaire Central System Cooling Coil



Sketch of a Frigidaire Central Cooling System

Frigidaire Central System Cooling Coils. Cooling and heating coils of this class are available in numerous shapes and sizes. The coils have extensive finned areas for providing contact with large volumes of air in relation to the size of the units. These coils are specifically designed for installation in the cooling and heating chambers of central systems. They can be used with existing warm air systems or with systems providing general ventilation.

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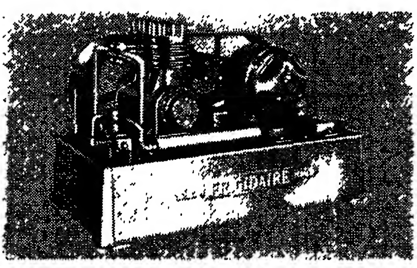

Frigidaire Four-Cylinder Compressor, Model FW-7500

Frigidaire Compressors. These are made in many sizes from small to large capacity. Each Frigidaire compressor is built like the finest automobile engine, as would be expected of a General Motors product.

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Complete data will be gladly furnished by Air Conditioning Division, Frigidaire Sales Corporation, Dayton, Ohio.

General Electric Company

AIR CONDITIONING PRODUCTS

Air Conditioning Department, 570 Lexington Ave., **New York, N. Y.**

For Steam, Hot Water, Vapor, or Warm Air Heating Systems, a complete line of Air Conditioning Equipment:

The G-E Oil Furnace (2 sizes) Type LA4 and LA5—See Page 666.

The G-E Winter Air Conditioner for Warm Air Systems (Type AA3)—This Air Conditioner, combined with the G-E Oil Furnace, provides all the four functions of winter air conditioning—heating, humidifying, filtering and circulation. A fan draws air from the return duct system through dry steel-wool filters, delivering it over the steam heating surface and humidifying screens, to the delivery duct system. Humidifying water is heated by an indirect heating coil, flow being controlled by a magnet valve operated from a Humidistat. Cooling coils for summer air conditioning optional.

Specifications

Output—130,000 B.t.u. per hour.

Power Input—260 watts at full load.

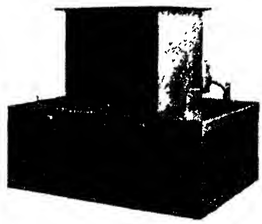
Humidifying Capacity—1.5 gal. per hour maximum.

Blower—Multi-vane, maximum delivery, 1600 c.f.m.

Heating Surface—$2\frac{1}{2}$ in. tappings for supply and return, 15 lbs. maximum operating pressure.

The G-E Winter Air Conditioner for Radiator Systems (Type AC1)

The Type AC1 air conditioner combines means for humidifying, filtering and circulating. It is mounted in the floor, either operating as an open system, or with return duct. Hot water for humidification is supplied from an indirect water heater connected to the heating boiler. Air is admitted through filters located on each side, discharging over the humidifying screens and finally through the delivery grille into the room. Control of humidification is obtained by means of a magnet valve operated by a humidistat.



Specifications

Fan—400 c.f.m.

Humidifying Capacity—$1\frac{1}{4}$ gallons per hour.

Power Input—45 watts.

The G-E Room Air Conditioner

The G-E Room Air Conditioner (2 sizes) provides complete year round conditioning—heating, humidifying, filtering, ventilating and circulating in winter—and cooling, dehumidifying, filtering, ventilating and circulating in summer. The heating surface is used with existing steam system. Automatic control of temperature is by means of a thermostatic inlet valve.

Ventilation is controlled by means of a damper in the outdoor air duct, and circulation is controlled by varying the speed of the blower motor. Humidifying is controlled by means of a switch. On the small size, the condensing unit for cooling may be mounted either integral or remote, while on the large size, the condensing unit is always mounted remote.

Specifications

Type	AD1	AD2
Ventilation	0–200 c.f.m.	0–200 c.f.m.
Recirculation	200–400 c.f.m.	400–600 c.f.m.
Heating Capacity—		
Low Speed	7900 B.t.u.	9200 B.t.u.
Medium Speed	12200 B.t.u.	16000 B.t.u.
High Speed	16800 B.t.u.	22000 B.t.u.
Cooling Capacity—		
Nominal Maximum	8500 B.t.u.	14000 B.t.u.
Humidifying Capacity	1.2 lbs. per Hr.	1.2 lbs. per Hr.

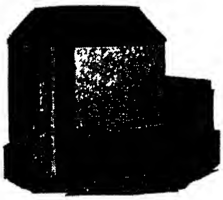
The Meyer Furnace Company

Peoria, Illinois

Manufacturers of Domestic Heating and Air Conditioning Units for Coal, Gas and Oil Burning

Branches and Distributors

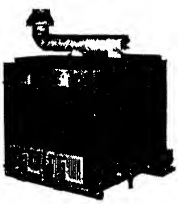
Kansas City, Mo.
Omaha, Neb.
Green Bay, Wis.
Pittsburgh, Pa.
New Orleans, La.
Detroit, Mich.
St. Louis, Mo.
San Francisco, Calif.



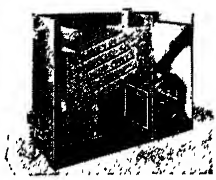
The WEIR Conditioned-Air Unit for coal or oil burning, built around the famous Weir Steel Furnace, is, as its name implies, a complete outfit for conditioning the air in the home during the heating season with respect to temperature, humidity, air-cleansing and positive circulation. Casing of insulated asbestos composition with pleasing red finish. Equipment includes automatic humidifier (which may be provided with room control if desired), renewable filters, centrifugal blower and fully automatic damper and blower controls. (Refer to table below for data on gravity as well as mechanical installations).

The WEIR Furnace

No.	Grate Surface (Sq Ft.)	Ratio Htg. to Grate Surface	Smoke Outlet Diam. (In.)	Gravity Circulation: Casing Dimen. Round (In.)	Gravity Circulation: Casing Dimen. Rect'lar (In.)	Gravity Circulation: Rated Output At Reg. (Btu/Hr.)	Gravity Circulation: Rated Output Pipe Area (Sq. In.)	Fan Circulation: Casing Dimen. (In.)	Fan Circulation: Air Delivery (C.F.M.)	Fan Circulation: Rated Output at register (Btu/Hr.)
621	1.26	41 2	9	48	. .	54,400	400	. .	....	....
624	1.78	33 9	10	52	47x50	73,600	541	47x90	1200	92,000
628	2.32	29.2	10	54	50x52	94,100	692	50x99	1600	118,000
630	3.08	26.4	10	58	54x56	119,000	875	54x103	2000	148,000
633	3.82	22.7	10	65	56x64	138,000	1015	56x110	2300	172,000
636	4 74	19.4	10	67	56x66	160,000	1180	56x118	2700	200,000
540	6.25	19.3	12	..	.....	....	....	60x106	4000	264,000
544	7.60	18 5	12	..	..	....	....	64x114	5000	316,000



The MEYER Gas Fired Air Conditioner automatically provides completely conditioned air for the heating season. It is as attractive in appearance as it is compact and efficient. Heating section of special heavy gauge welded steel, gas- and fume-tight construction; casing of insulated asbestos composition with pleasing red finish. Equipment includes specially designed automatic humidifier (which may be provided with room control if desired), renewable filters, centrifugal blower and fully automatic controls. Manufactured or natural gas.



Data on refrigerating equipment for summer cooling, to be used in conjunction with any of the mechanically-circulating systems described, will be supplied upon request.

No.	Output at Bonnet (Btu/Hr.)	Input at Burner (Btu/Hr.)	Vent Diam. (In.)	Dimensions W. (In.)	Dimensions L. (In.)	Dimensions H. (In.)	Air Delivery 1/4" S.P. (C.F.M.)	Motor Size (HP.)
MEYER Gas-Fired Air Conditioner								
1-B	72,500	90,000	4	20	53	42	1000	1/6
1½-B	108,000	135,000	5	27	60	44	1500	1/4
2-B	145,000	180,000	7	40	53	42	2000	1/3
3-B	217,500	270,000	8	60	53	42	3000	1/2
4-B	290,000	360,000	9	80	53	42	4000	3/4
5-B	362,500	450,000	10	100	53	42	5000	3/4
MEYER Gravity Gas Furnace								
C-100	80,000	100,000	5	38	38	60		
C-120	96,000	120,000	6	42	42	67		
C-150	120,000	150,000	6	42	42	67		

The MEYER Gravity Gas Furnace

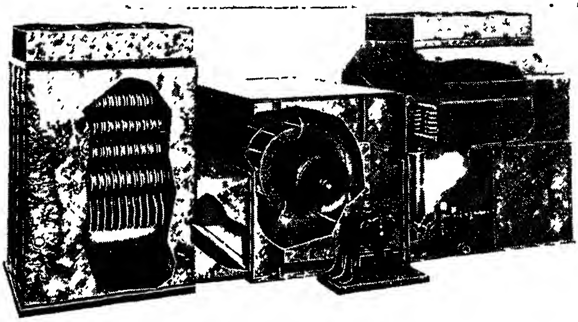
Designed for high efficiency and resultant fuel economy with gravity circulation. Heating section of all-steel, welded construction with unusually large vertical heating surface. Casing same as on Air Conditioner. Complete with automatic controls. Manufactured or natural gas. Priced for the pocketbook of the average home.

L. J. Mueller Furnace Co.

ESTABLISHED 1857

2009 W. Oklahoma Avenue, **Milwaukee, Wis.**

CLIMATOR AIR CONDITIONING SYSTEMS



Climator Year Round System

In the illustration is shown a Mueller Climator installation designed to handle all functions of both winter and summer air conditioning.

During winter, the circulated air is warmed by a heating unit specially designed for the fuel to be used, coal, oil or gas. Controlled humidity is supplied by the air washer. Removal of air-borne impurities is accomplished by the filters. Adequate circulation, uniform temperature and silent air distribution is secured by the fan.

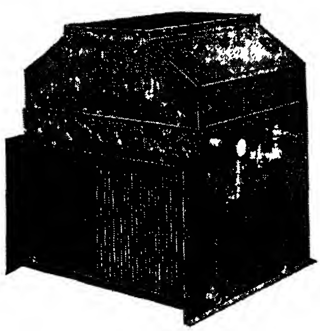
In summer, temperature reduction and dehumidification is secured by Frigidaire refrigerating equipment, the cooling coils being enclosed within the washer housing. Compressor may be located adjacent to the equipment, or in any desired location.

Where cooling and dehumidification is omitted, arrangement may be such that it may be added at a later date without change in equipment. If no provision for refrigeration is desired, a somewhat smaller washer is employed, and a more compact assembly may be secured.

The air conditioned in the Climator system is distributed to all rooms through ducts and discharged through small overhead registers. Individual returns from each room secure perfect balance.

Catalog and detailed information furnished upon request.

Climator III Unit



Climator III Unit

This unit is a fan, filter and washer in a single casing finished in green prismatic lacquer. It may be connected to any type warm air furnace, new or existing installations. It will pass through an opening 27½ in. wide. The return air duct, recirculating air from the rooms, connects directly to the unit, the only method by which an efficient, balanced installation can be secured. Fan is of adequate capacity to secure positive air delivery against resistances encountered through ducts, washer and filters. Made in one size only, with rated c.f.m. to 2,250, and adjustable speed driving pulley for adjustment to requirements.

Parks-Cramer Company

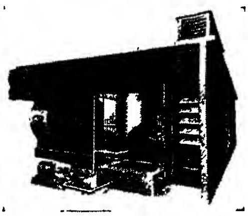
Fitchburg, Mass. **Charlotte, N. C.**

CERTIFIED CLIMATE

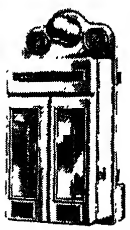
Complete Air Conditioning Systems including Heating, Cooling, Humidifying, De-humidifying, Ventilating, Refrigeration, Air Filtering and Air Washing

AUTOMATIC REGULATION

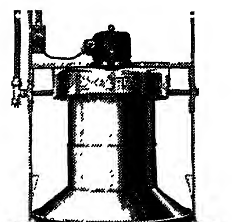
Industrial Heating by Oil Circulation with Merrill Process



Central Station



Psychrostat



High Duty Humidifier

Mistyfier

Industrial Air Conditioning

Helps in many industries, notably, Textiles (Cotton, Wool, Worsted, Silk, Rayon, Jute); Printing and Lithographing; Cigar, Cigarette and Tobacco; Clothing; Paper and Envelope; Leather and Shoes; Wood Products; Cereals; Storage of Perishables; Ceramics; Celluloid; Glassine Paper; Starch and Dextrine; Cement. Installations similar in design are effective in Hospitals, Art Galleries, Auditoriums and Restaurants.

Automatic Regulation

The Psychrostat for accuracy, durability, sensitivity. Hygrostat (not illustrated) where requirements are not so exacting. Psychrostat uses the Principle of the Sling Psychrometer. U. S. Government uses Sling Psychrometer in all Weather Bureau Stations. An Air Conditioning System is no better than its Regulation.

Central Station Air Conditioning

A centrally located apparatus supplying maximum moisture needed with positive pre-determined air change. Usually includes indirect radiation for heating—may include refrigeration and cooling. All air is washed. Automatic Regulation essential. While initial and operating costs are high, this absolute and centralized control of Certified Climate is often desirable;—sometimes essential.

High Duty Humidifier

Water under pressure generates spray. Excess water returns to filter tank and recirculated. Evaporation per unit high; two sizes of heads each with two sizes of nozzles give flexible capacity for varying conditions. Circulation increased by individual motor-driven fan. Spray thoroughly diffused and distributed over wide area.

Centrifugal Humidifier (Not Illustrated)

Water without pressure. Fine spray.

The Mistyfier

A mechanical humidifier made according to same high standards as Parks Certified Climate devices but refined for home and office use. Quiet and automatic—not a toy. Inexpensive to operate;—wastes no water. Uses little electric current. Permanently though flexibly connected to water supply without expensive plumbing. Safeguards health. Increases comfort. Raises effective temperature during heating season, lowers it during dry non-heating season.

Unit Air Conditioner (Not Illustrated)

Principle similar to Mistyfier but for larger spaces, or may be made part of basement installation.

Niagara Blower Company

AIR ENGINEERING EQUIPMENT AND SYSTEMS

General Sales Office: 6 East 45th Street, **New York City**

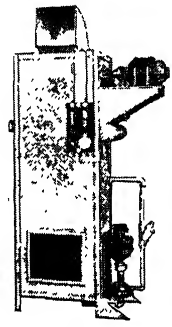
BUFFALO BOSTON PHILADELPHIA CLEVELAND
PITTSBURGH CHICAGO SEATTLE SAN FRANCISCO

PRODUCTS—*Air Conditioning, Humidifying, Dehumidifying,* Comfort Systems, Niagara Air Conditioners, Niagara Fan Coolers, Niagara Fan Heaters, Niagara Aluminum Cooling Coils, Heating Coils.

NIAGARA

NIAGARA AIR CONDITIONING SYSTEMS

For human comfort and for all industrial applications requiring controlled climatic conditions of temperature, relative humidity, air purity and air movement.

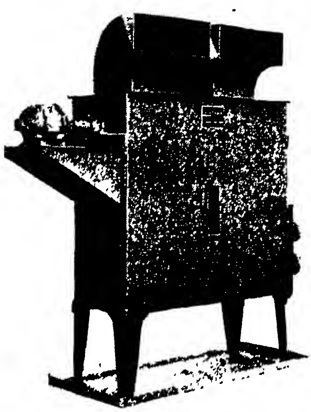


NIAGARA

Niagara Air Conditioner Type A

Illustration of single fan unit. Also manufactured in 2-, 3- and 4-fan units and in 7 sizes.

Niagara Surface Cooling Method



Niagara Fan Cooler
Manufactured in 1-, 2-, 3- and 4-fan units and in 7 sizes.

Air conditioning for human efficiency and comfort. A year 'round operating system providing winter heating and humidifying and summer cooling and dehumidifying for offices, stores, restaurants and all places where people gather. An advanced engineering method introducing simplified apparatus and controls superior in operating results, in economy and longer life.

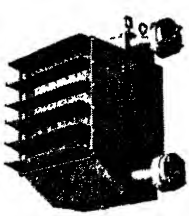
Niagara Air Conditioner —Type A

Maintains constantly, or makes any change required in temperature and relative humidity; dries or moistens within tolerance of 1 deg. F. and 2 per cent R. H. in processing hygroscopic materials; cleans air most effectively; secures saturation for dehumidifying. Seven sizes.

Niagara Fan Cooler

Recommended for comfort cooling, process cooling, low temperature storage for dairies, fruits, meats, food products, fur storage vaults, etc.

Gives complete circulation of air at desired temperature with even temperature at all points. Manufactured in seven sizes.



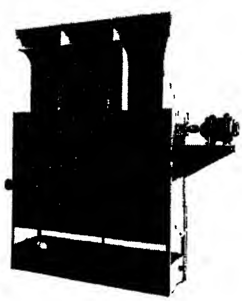
Niagara Disk Fan Cooler
Manufactured in 7 sizes including 2-fan unit.

Niagara Disk Fan Cooler

For overhead suspension, saves space and provides the efficiency of moving air cooling for small storage areas, market coolers, etc. Seven sizes.

Niagara Humid Heater

Recommended for industrial applications where heat and humidity are required as in manufacture of textiles, cordage, printing and paper-converting plants.

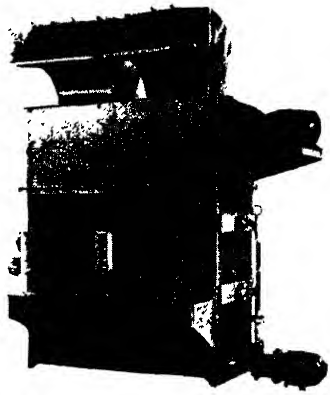


Niagara Humid Heater

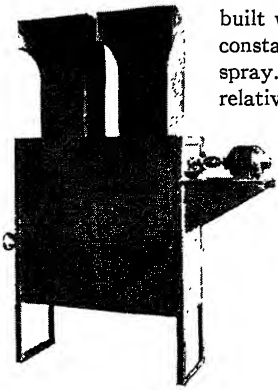
Niagara Spray Cooler

For food product applications, especially meat chilling and storage pre-cooling and storage of fruits and vegetables. Prevents drying out, wilting.

Niagara Spray Cooler is built with cooling coils in a constant brine or water spray. Maintains constant relative humidity as required.



Niagara Spray Cooler
Illustration of two-fan unit. Also manufactured in 1-, 3- and 4-fan units and in 7 sizes.



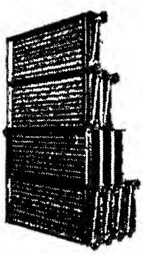
Niagara Fan Heater
Illustration of 2-fan unit. Also manufactured in 1-, 3- and 4-fan units and in 7 sizes.

Niagara Fan Heaters

For the heating and ventilating of large areas, Niagara Fan Heaters put the heat immediately where needed in the working zone, give quicker heating up to working temperatures. Definitely built to the highest possible standards; welded aluminum heating coils, welded frames.

Niagara Aluminum Cooling Coils

Tested cooling coils are used in Niagara Fan Coolers and Niagara Surface Method air conditioning. Encased, seven standard sizes for blast cooling installations. 20-in. and 30-in. widths. Aluminum coils in aluminum cases.



Niagara Aluminum Heating Coils

Niagara Aluminum Heating Coils

For use with fan heating systems giving the advantage of aluminum, light weight and resistance to corrosion. Manufactured in two widths, 20-in. and 30-in., and in two types of three lengths, giving a complete range of sizes. 150 lbs. working steam pressure.

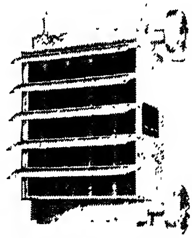
Niagara Aluminum Booster Heaters are used for reheaters to control room temperature independently of fan system.

Niagara Aluminum Booster Heater

Write for Bulletin No. 20

Niagara Disk Fan Heaters

A most effective suspended heater. Operates with lower discharge temperature, cuts down roof and wall losses.



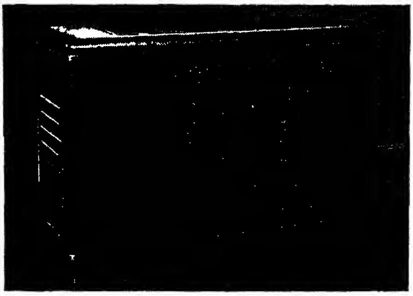
Niagara All Aluminum Disk Fan Heater

Westinghouse Electric & Manufacturing Co.

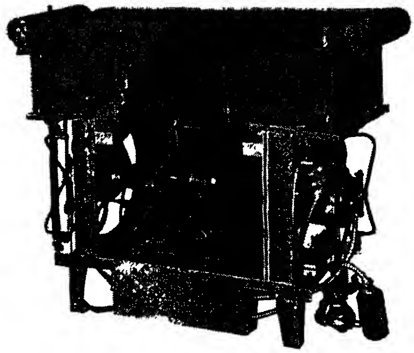
East Pittsburgh **Pennsylvania**

Sales Offices and Service Shops in over 110 Cities

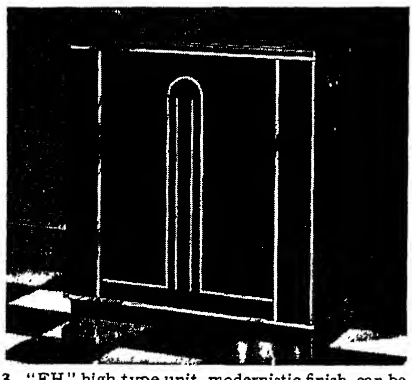
UNIT AIR CONDITIONING EQUIPMENT



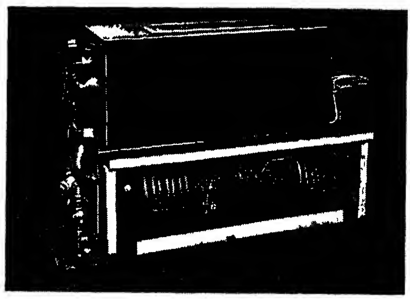
1. Westinghouse "EL" low type air conditioning unit for stores, offices, homes, etc. Walnut finish cabinet. Heats and humidifies in winter, cools and dehumidifies in summer, and filters and circulates air the year 'round.



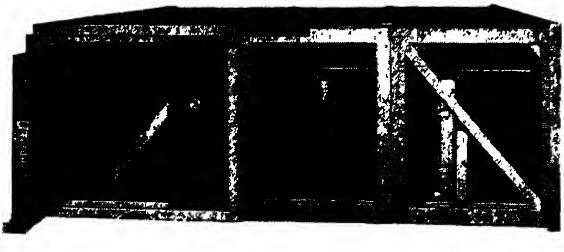
2. Outer cabinet removed, showing finned type coils for heating, cooling and dehumidifying. Air is drawn in by fans at each end and deflected upward by baffles. A water spray, atomized in air stream, humidifies winter air. Installed in room, with refrigerating unit in basement, adjoining room or closet.



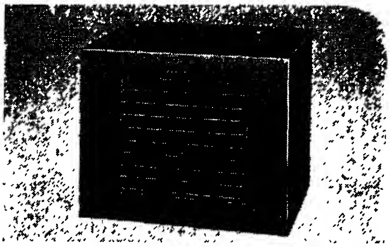
3. "EH" high type unit, modernistic finish, can be installed in aisles, beside columns or along the wall. Both high and low models are made in both finishes. Rated at 12,000 B.t.u. per hour cooling and 24,000 B.t.u. per hour heating.



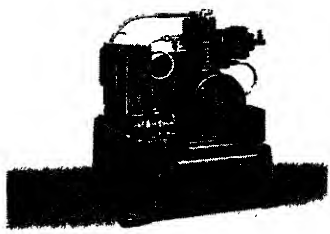
4. Self-contained mobile unit with cover removed, showing refrigerating unit in lower half, with evaporator above. Cooling and dehumidifying only, rated at 6,000 B.t.u. per hour. Quiet in operation and easily installed. Can be equipped with wheels and moved from room to room.



5. Suspended type air conditioning unit type ES-62. Can be suspended from ceiling or mounted on the deck over show windows. Used with or without ducts. Summer and winter operation.

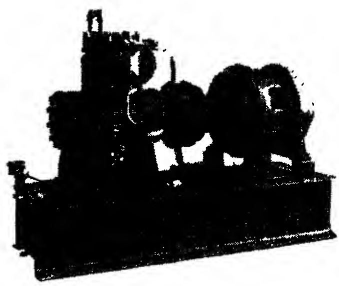


6. Small suspended unit, type ES-10 for summer operation. Cools, dehumidifies and circulates the air. Adjustable louvres direct air stream as desired.



7. Refrigerating unit, smallest size, rated at 12,000 B.t.u. per hour. Many sizes available, air and water cooled for all commercial electrical circuits.

8. Larger size refrigerating unit type RW-6, water cooled. All Westinghouse refrigerating units have multi-cylinder, vertical, single acting compressor direct-connected to electric motor, are designed specifically for unit system of air conditioning, and for use with Freon, the new non-poisonous, non-inflammable, non-irritant refrigerant. There are no valve cams or levers, and crankshafts are dynamically balanced. The units are compact, efficient and quiet.



The Westinghouse *Steam Jet Refrigeration Unit* (not shown) utilizes steam pressure to provide cooling effect for air conditioning or water cooling. It is highly economical in operation and maintenance because it has few moving parts.

AIR CONDITIONING UNITS A-C. OR D-C.

Model	Type	†Operation	Dimensions, Inches						Weight, Lbs.		Air Discharge Cu. Ft. Min.	Motor Horse Power
			With Case			Without Case			Complete	Without Case		
			Lgth	Hgth	Dpth	Lgth	Hgth	Dpth				
Floor—Low	EL-12	S W	39	27	13½	34	23	12	226	195	450	1/60
Floor—Low	EL-10	S	39	27	13½	34	23	12	180	150	450	1/60
Floor—High	EH-12	S W	39	40	13½	34	38	12	272	220	450	1/60
Floor—High	EH-10	S	39	40	13½	34	38	12	225	175	450	1/60
Suspended	ES-10	S	23½	21	13¾	22	20	12	136	120	450	1/60
Suspended	ES-22	S W	31½	20	54	..	..	..	425	...	900	1/4
Suspended	ES-20	S	31½	20	38¼	..	..	..	375	...	900	1/4
Suspended	ES-42	S W	54	20	54	..	..	..	850	...	1800	1/2
Suspended	ES-40	S	54	20	38¼	..	..	..	600	...	1800	1/2
Suspended	ES-62	S W	65	22½	59¾	..	..	..	1200	...	2700	3/4
Suspended	ES-60	S	65	22½	44	..	..	.	870	..	. .	..

†S for summer, W for winter.

REFRIGERATING UNITS

Type	Cooling	**Capacity B.t.u./hr. Cooling	Electric Power	Outside Dimensions, Inches			Wt., Lb.	Motor H.P. (Approx.)	Charge	
				Lgth	Wdth	Hgth			Refrigerant, Lb.	Oil, Pints
RW-1	Water	12000	50 or 60 cy.	19¾	19¾	23½	350	1½	13	4½
RA-1	Air	12000	50 or 60 cy.	28	19¾	26½	400	2	19	4½
RA-1	Air	12000	DC or 25 cy.	46	20¼	24½	450	2	13	4½
RW-2	Water	24000	50 or 60 cy.	20⅞	20¼	29¾	450	3	18	5
RA-2	Air	24000	50 or 60 cy.	37	24	32	500	3	25	5
RA-2	Air	24000	DC or 25 cy.	52	24¼	29¾	575	3	18	5
RW-4	Water	48000	DC, 60, 50, 25 cy.	58½*	23½	38½	AC-1085 DC-1225	5	40	13
RA-4	Air	48000	DC or 60 cy.	76¾	33½	32⅜	....	7½	40	13
RA-4	Air	48000	50 or 25 cy.	76¾	33½	32⅜	....	6½	40	13
RW-6	Water	72000	DC, 50 or 60 cy.	58½*	23½	38½	AC-1085 DC-1225	7½	40	13
RA-6	Air	72000	DC or 60 cy.	76¾	33½	32⅜	AC-1850 DC-1975	10	40	13
RA-6	Air	72000	50 cy.	76¾	33½	32⅜	1850	8½	40	13
RW-12	Water	144000	25, 50 60 cy. or DC	Refer to nearest Westinghouse Air Conditioning Dealer for information on these ratings.						
RW-18	Water	216000	25, 50 60 cy. or DC							

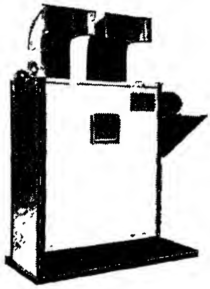
*Add 4-in. for DC motor. **Based on 40° evaporator temperature and with 60 cycle or DC motors. Capacities with 25 or 50 cycles will be lower.

York Ice Machinery Corporation

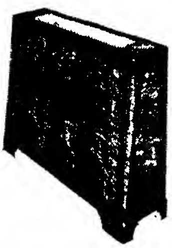
General Offices: **York, Pennsylvania**

Direct Factory Branches in 71 U. S. Cities

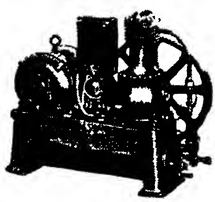
Complete Air Conditioning and Refrigerating Systems for maintaining proper atmospheric conditions for industrial processes and human comfort. Available in central and unit systems . . . from fractional tonnage up to any capacity required.



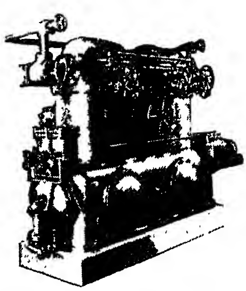
Coil Type Air Conditioner



Floor Type Unit Air Conditioner



Freon (F-12) Refrigerating Unit



Vertical Single Acting Freon (F-12) Compressor

Coil Type Air Conditioner:

A self-contained air cooling unit provided with coils for direct expansion of the refrigerant or for circulation of water or brine. Also furnished for ceiling mounting. Coil design insures maximum efficiency. Low speed fans designed for quiet operation. Adapted to thermostatic control with defrosting feature.

Spray Type Air Conditioner:

Complete self-contained air conditioning unit requiring minimum space and including air washer with refrigerating coil, air heating coils, water heating coils, fan and motor, pump and motor, temperature and humidity controls, all assembled as a single unit.

Floor Type Unit Air Conditioner:

A compact, year 'round, unit air conditioner for office, home, restaurant, store . . . . wherever capacity or building requirements dictate its use. Provides summer cooling and dehumidifying; winter heating and humidifying. Attractive lines and finish permit it to harmonize with any type of furniture or decorations.

Air Washer:

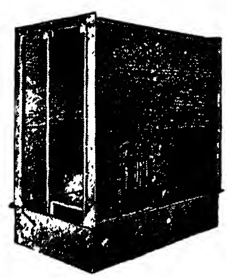
Galvanized iron and copper air washers of extra heavy construction, designed for air conditioning or ventilating duty with water or brine. Washers furnished with or without cooling coils. Adjustable self-cleaning mist nozzles insure maximum humidifying efficiency.

Refrigerating Systems for Air Conditioning:

Complete refrigerating systems for use with Freon (F-12), Ammonia and Carbon Dioxide. Because refrigeration for air conditioning is essentially a water cooling problem, York has developed properly balanced, standard water cooling systems for this duty. Designed especially for human comfort applications, Freon was selected as the most suitable refrigerant because of its outstanding characteristics.... odorless, non-toxic and non-poisonous, non-inflammable and non-explosive, non-irritant and non-corrosive. Freon's thermal properties make it ideally suited for use in vertical single acting reciprocating compressors, the standard of simplicity and efficiency.

York Engineering Service:

Because of York's vast fund of technical information and operating data in the field of air conditioning and refrigeration, York engineers can offer many helpful suggestions in the solution of your problem. Architects, Engineers, Contractors and others are invited to avail themselves of this service.



Air Washer

Knowles Mushroom Ventilator Co.

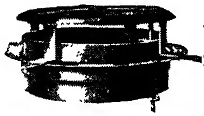
41 North Moore Street, New York

Knowles Air Diffusers for Auditoriums of Theatres, Churches, Schools

ORIGINAL——RELIABLE——STANDARD

Giving Satisfactory Service in Over 4,000 Installations

PRODUCTS:—Cast-Iron and All-Steel Adjustable Mushroom Ventilators; Aisle Hood, Tu-Way Air Deflectors; Round and Oblong Gallery Riser Vents

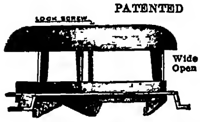


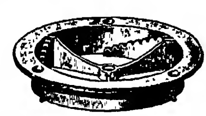
Aero Valve—The Improved Mushroom Air Diffuser

The newest Knowles product. An up-to-date mechanically correct air unit of fixed height made of HEAVY steel, with strong supports bearing across 1-in. flange. Easy to regulate and install and low in price. Combines high efficiency with convenient adjustment affording finest control of air by simply turning screw on TOP of CAP. Can be positively locked at desired adjustment by side screw—is noiseless and made so that it cannot be taken apart before or after installation—an exclusive feature. Quick Anchorage to Wood or Concrete floors in one piece—Made in 6-in. and 8-in. diameter sizes—Same capacities as **Nu-Notch** type (see table below).

Knowles *Nu-Notch* Mushroom Ventilator or Air Diffuser

PATENTED



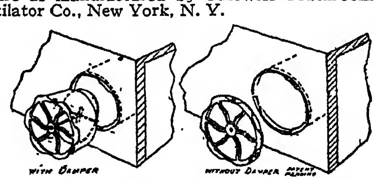


The ideal cast-iron mushroom. Head has three outer bearings and is absolutely rigid. Ten recessed notches give close regulation of air. Locked into positive adjustment by tightening head screw with key—a new feature to prevent tampering by unauthorized person. Supplied with dome or flat tops. Three screw holes in floor flange for fastening to wood, or three angle L lugs for setting in concrete, or three tapped holes in floor collar for fastening with set-screws to sleeve.

Size	Cu. Ft. per Min. at 300 ft. Vel.	Area, Sq. Ft.	Weight Lbs.
5″ diam.	42	0.1364	3.50
6″ "	60	0.1964	4.25
7″ "	81	0.2673	5.75
8″ "	105	0.3491	8.00
10″ "	165	0.5454	11.75

Specifications—Furnish and install where indicated on drawings or as hereinafter specified (5-in.), (6-in.), (7-in.), (8-in.), (10-in.), (Dome Top), (Flat Top), **Nu-Notch** Cast-Iron Mushroom Air Diffusers with recessed notches for the permanent adjustment of mushroom caps at any desired opening together with center locking screw feature as manufactured by Knowles Mushroom Ventilator Co., New York, N. Y.

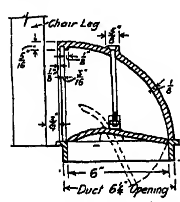
Knowles *Disc-Loc* Gallery Riser



Ventilators are designed to insure better control of air. Round holes in the gallery risers do not weaken the construction but, on the contrary, with the cast-iron rings it is actually strengthened. The cast-iron grille is quickly inserted and locked by a simple twist motion—no bolts, screws or springs. Furnished with or without damper to fit same rings. MADE IN FOUR SIZES (See Booklet).

Tu-Way **Air Deflectors** are designed to deliver maximum area with minimum fixed height. The air is discharged at *both sides along the row of seats.* Heavy Cast-Iron — Ornamental design — Adjustable double-wing curved damper deflects air downward. Fine, positive adjustment—Rattleproof, noiseless. A DEVICE THAT DELIVERS THE FULL AREA OF FLOOR OPENING. Floor opening 6½ by 8½ inches—Area ⅓ Sq. Ft. —100 CFM at 300 Velocity.

PATENTED



Standard Aisle Hood (Cast-Iron) Air Deflectors are used to throw the fresh air out into the aisles in one direction. A curved damper reduces friction loss.

Large size: 8 in. long, 6 in. wide, 6 in. high; area ⅓ sq. ft.; Small size: 8 in. long, 4½ in. wide, 4½ in. high; area ¼ sq. ft.

Send for New Booklet of Complete Engineering Data or Samples

AMERICAN AIR FILTER COMPANY INC.

INCORPORATED

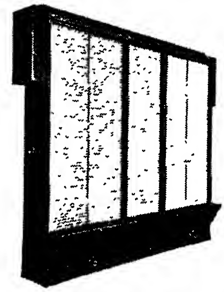
1st Street and Central Avenue, **Louisville, Ky.**

Representatives in Principal Cities

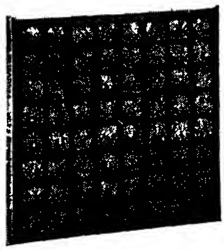
Dust Engineering—Dust Engineering is that branch of applied science which deals with the origin, nature and characteristics of the small solid air-borne particles called "dust," and the development of methods, processes and apparatus for its control or elimination.

The American Air Filter Company, Inc., has had an important part in advancing the science of Dust Engineering. The efforts of its Research and Engineering Staff for the past twelve years have been devoted exclusively to the study of dust problems and the development of a complete line of air cleaning equipment for modern air conditioning, building ventilation and the control of process dust in industry.

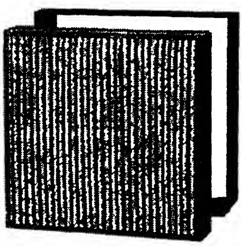
American Air Filter products, therefore, not only embody the knowledge accumulated from years of constant research and the experience gained from designing, building and applying thousands of air filters, but are backed by ample technical and financial resources to insure their outstanding position in the Dust Engineering field.



Automatic Multi-Panel Filter

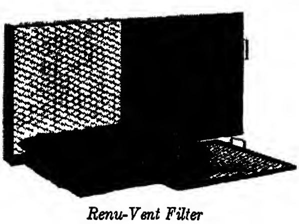


Throway Air Filter



Airmat Drifilter Type K

Products—American Air Filters are available for every condition, with operating characteristics and efficiencies to suit specific problems. In general, there are two distinct types based upon the "viscous film" and "dry mat" principles. Each type is made in several styles which differ in method of operation, servicing, space required and initial cost to meet the various conditions encountered in air cleaning problems. A discussion of various filter types will be found in the Technical Data Section under "Air Cleaners."



Renu-Vent Filter

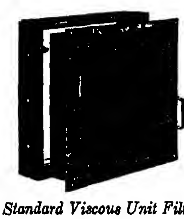
Air filters are generally used for the removal of dust, dirt, bacteria and other foreign matter from the air and are applied to general ventilation, modern air conditioning, process dust control; for air compressors and Diesel Engines; mill motors, turbo-generators and other electrical applications; and for air or gas under pressure to remove entrained oil, moisture and dirt.

Air Filters in Air Conditioning—Filtered air is today recognized as essential in modern air conditioning. There are other important factors which contribute to our comfort such as temperature, air movement and humidity, but science today emphasizes the prime necessity of pure air for health and efficiency.

Air cleaners have, of course, always been considered an integral part of large central systems. These are usually of the fully automatic type such as the Multi-Panel filter, illustrated in the accompanying photograph.

There are now available to manufacturers of unit air conditioners moderate priced unit filters, such as the Re-Nu filter, the Drifilter and the Throway filter, illustrated herewith.

The Re-Nu filter is an entirely new departure in air filter construction. It consists of a permanent metal frame provided with



Standard Viscous Unit Filter

a removable cover and renewable filter pad. The cover is easily removed without the use of tools, and filter pad can be lifted out and replaced with a new one at very small expense in less than a minute's time.

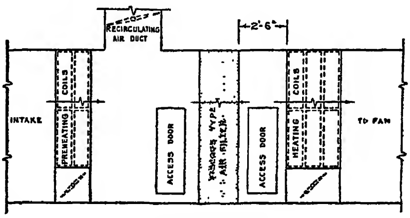
The Drifilter consists of the Airmat filtering media mounted on a supporting member and arranged in saw-tooth fashion, as illustrated. The filter media is reasonable in price and can be easily replaced when desired.

The throway filter, as the name implies, is an inexpensively constructed unit designed to be discarded after it has served its maximum period of usefulness and replaced with a new filter unit. The filter pad is enclosed in a perforated cardboard container which makes it possible to dispose of the dirty filter by burning it in a furnace or incinerator.

There is probably no single item which costs as little and may mean as much in the design of an air conditioner as air filtration. These units are furnished in any dimensions or shapes desired. They are usually built in units handling 400 c.f.m. and from 2 in. to 4 in. thick. They are usually made in the following sizes—20 x 20 in., 16 x 25 in. and 16 x 20 in. Cleaning efficiencies from 90 to 99 per cent can be secured, with a resistance to air flow ranging from $\frac{1}{16}$ in. to $\frac{3}{8}$ in. water gauge.

Automatic Self-Cleaning Air Filters—The American line of automatic air filters is among the most complete that has ever been offered, the most popular types being Multi-Panel, Horizontal and Phoenix Filters.

All types are furnished for either continuous or intermittent service and are available in sizes and set-ups suitable to



Typical Installation

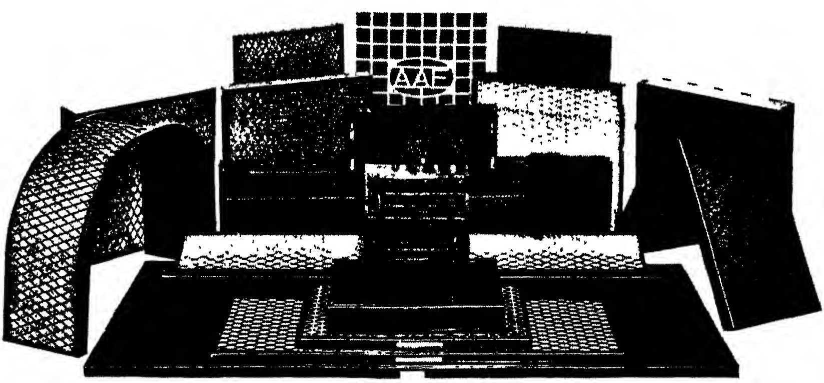
any desired capacity or space condition.

Standard Viscous Unit—The American Unit Air Filter incorporates the time tested unit principle of construction. Each unit consists of a standard steel frame and interchangeable cell equipped with automatic latches to facilitate removal for cleaning and recharging.

Airmat Filter Dry Type—The filtering media in this type is the Airmat sheet, a dry filter mat composed of thin sheets of gauzy, cellulose tissue. The Airmat sheets are supported in screen pockets mounted in a unit frame of box-like construction. These unit frames can be set up to meet any capacity requirement or space condition. The Airmat sheets are renewable —their life depending on the dust condition and hours of daily service.

Airmat filters are used both for air conditioning and industrial air conditioning. In the latter field they are particularly well adapted for the recovery of valuable dusts and for abating the dust nuisance which confronts so many industrial plants.

Our standard data books and catalogues are to be found in most engineering files or libraries. We will be glad to furnish full data to engineers or manufacturers interested in this subject.



Various Types of Unit Air Filters for Air Conditioning Work

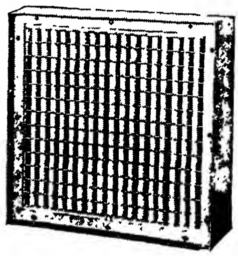
Coppus Engineering Corporation

339 Park Avenue, **Worcester, Mass.**

MANUFACTURERS OF AIR FILTERS, STEAM TURBINES, FORCED DRAFT BLOWERS, COOLING FANS

Annis Unit Filter

The Annis Unit Filter is of the dry type, using a removable filter glove of special wool felt supported by a rigid welded wire frame and held tautly over the wire frame by a spreader grid the edges of which are extended box-like to give protection to the filter element. All metallic parts are rust-proofed (Bonderized, Cadmium plated), the steel filter box Duco painted.



Clean Air Side of Stationary Unit Filter

Specifications

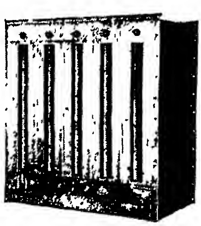
Normal Rating: 800 c.f.m.
Resistance when clean: .2 in. W.G.
Dust Arrestance (cleaning efficiency): for dust particles of 10 micron size (0.0004 in.) and larger 99¾%+.
Dimensions: 20 in. by 20 in. by 6½ in.
Weight per unit: 25 lbs.

Cleaning Filter Elements with Portable Vacuum Cleaner

Outstanding Advantages

1. It has an exceptionally high dust arrestance.
2. It maintains a high dust arrestance even under diverse conditions of neglect.
3. Its operation is not impaired by atmospheric conditions.
4. It is a Medium Air Resistance Type (Class C) according to the A.S.H.V.E. Code for Air Cleaning Devices.
5. It is easily and quickly cleaned without removing the filter element.
6. Its cost of upkeep is very low because the *permanent* filter element is reconditioned periodically with a vacuum cleaner.
7. It combines scientific knowledge and practical engineering methods with highest quality of material and workmanship.

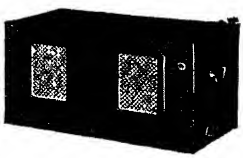
Automatic, Self-Cleaning Air Filter



It uses the same special wool felt as filter medium arranged in the shape of an endless belt in zigzag fashion over rolls. Either by means of a time clock at predetermined intervals, or with the help of a sensitive pressure gauge, when a maximum air resistance is reached, the filter curtain is moved by a small geared motor over the rolls and passes over the nozzle of a small vacuum cleaner. This automatic cleaning operation takes place while the ventilating system is in operation and is finished within 15 to 20 minutes. The motion of the filter curtain is then stopped automatically. The only attention necessary is the occasional removal of the dust bag from the vacuum cleaner. Built in capacities from 3000 c.f.m. up.

Coppus Window Air Filter

It supplies a continuous flow of filtered air, is extremely quiet in operation, keeps out street noises, eliminates dirt and dust including more than 90 per cent of invisible particles, and last, but not least, is practically 99 per cent plus efficient against rag weed pollen in concentrations commonly found in the hay fever season. Its use is recommended for offices, homes and hospitals.



Air Filters for Compressors and Internal Combustion Engines.

Steam Turbines, Horizontal and Vertical, 1 to 60 Hp.

Forced Draft Blowers.

Portable and Cooling Ventilating Fans.

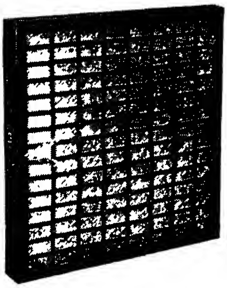
Owens-Illinois Glass Company

Industrial Materials Division

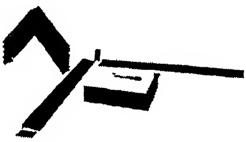
MANUFACTURERS OF DUSTOP AIR FILTERS

Toledo, Ohio

DUSTOP . . . the Glass Wool Air Filter of Highest Efficiency and Lowest Cost



The **Dustop** filter—standard size 20″x20″x2″. Rating—Capacity per unit, 800 c.f.m.—Maximum velocity recommended 300 f.p.m. —Resistance at rated capacity: Clean, .25″ w.g. (Two filters in tandem within unit frame)—Dirty, .36″—.40″ w.g.

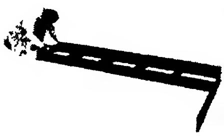


Dustop frames are not bulky, and are easily assembled—only a screw driver is needed. The frame units can be nested for handling and shipping, which effects considerable saving in freight, and greatly reduces labor necessary in carrying equipment to site and erecting it.

First assembly operation—Corner angle and side member are bolted to Base Plate.

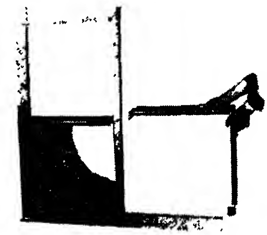
DUSTOP
GLASS WOOL AIR FILTER
SOLD BY
FAN AND AIR CONDITIONING EQUIPMENT MFGS.
FURNACE MANUFACTURERS
AUTHORIZED DISTRIBUTORS

With the side member at right angles to the Base Plate, the first "L" shaped frame member is bolted in the corner to both the side member and the Base Plate.

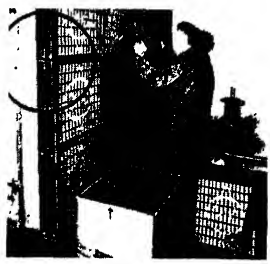


This first row of cells is completed by bolting one end of each frame member to the preceding one and the other end to the Base Plate. This is done so that the bolts through the Base Plate can be inserted easily.

The completed rows of units are raised into approximate position to complete assembly.



The **Dustop** frame assembles into a bank of any required number of units. To accommodate any specified volume of air two **Dustop** filters are inserted in series, for greatest economy and a cleaning efficiency from 96% to 98%.



The replacement operation is simple. Dirty filters are easily lifted from the frame and can be removed from the premises in the shipping carton in which the new ones arrived.

The **Dustop** glass wool air filter will clean the air in every type of commercial and industrial building at lowest cost. **Dustop** filters cost less for initial installation and less for replacement in maintenance. **Dustop** filters maintain an efficiency of 96% to 98% in removing not only dust, dirt, lint and soot from air, but also hay fever pollen, bacteria and other harmful impurities. Hundreds of **Dustop** installations are now in service efficiently cleaning air. **Dustop** filter banks are available through leading heating and ventilating supply houses and fan manufacturers everywhere. Replacement filters are available in all principal cities.

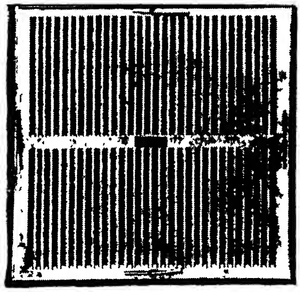
Staynew Filter Corporation

Air Filters for Buildings and Mechanical Equipment

6 Leighton Avenue, **Rochester, N. Y.**

Products—

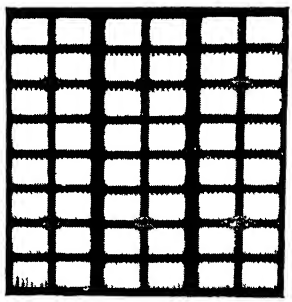
Protectomotor Dry Type Positive Filters for removing dust, dirt and foreign matter from small or large volumes of air at atmospheric or at higher or lower pressures. Made in various types and sizes for buildings, windows, oxygen chamber in hospitals, furnaces, pipe lines, air compressors, diesel engines, blowers, motors, pneumatic systems, air brakes, etc.



Protectomotor Building Filter

Operation—

The efficiency of Protectomotor Air Filters is exceptionally high due to their large filtering surface within a relatively small space. The intake air currents move parallel to the filtering surface at very low velocity, so that the dust and dirt are not packed on to the felt, but remain in a loose and porous condition, which permits the air to pass through the accumulated dust and dirt quite as readily as through the felt itself. Dust and dirt do not enter pores of felt, which is of extremely fine texture.



Multi-V-Type Building Filter

Efficiency—

Due to the low air velocity and the fine texture of the felt, practically complete removal of dust is obtained. An exceptionally high efficiency is maintained, even on fine air-floated dust. Efficiency is not greatly affected by continuous service, nor by any change in volume of air passed.

Pressure Drop—

Resistance to the flow of air is less than ¼ in. water gauge when operated at rated capacity. Less pressure drop, when required, may be obtained by using an oversize filter.

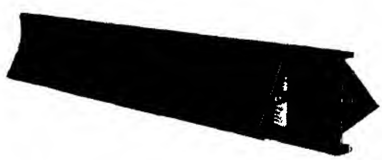
Cleaning—

Under ordinary conditions filters operate from six months to a year without attention. A special cleaning device enables the material collected on the filtering surface to be very quickly and completely removed by compressed air or vacuum cleaner. The material may be reclaimed, if valuable, without contamination, since no sticky or adhesive oils are used. No spare parts or cleaning tanks are necessary. Less than two minutes' time, per 1000 cu. ft. per minute of air capacity, required for cleaning. Filters may be cleaned while in operation, without removing filtering units.

Protectovent Window Ventilator
One filter cell removed, exposing one of the fans and motors

Application—

All filter units or filter assemblies are complete in weatherproof housing, ready to attach to air intake pipe for application to engines, compressors, etc. For building ventilation, equipment is easily adapted to space available.



Diagonal view showing adjustable ends storm hoods, filter and deflector plate. Note the large, active surface

The Vinco Company, Inc.

305 East 45th Street — New York, N. Y.

VINCO Boiler Cleanser

A positively harmless insoluble powder cleanser for new, remodeled and old heating systems. A unique, scientifically processed compound of ingredients, on a special formula not to be confused with other powder boiler cleaners.

ADOPTED BY

American Radiator Corp.
American Gas Products Co.
Barnes & Jones
Boynton Furnace Co.
**Burnham Boiler Corp.*
Gorton Heating Co.
Hart & Crouse Co.
Heggie-Simplex Boiler Co.
Hoffman Specialty Co.
Ideal Boiler Co.
International Boiler Works
Kewanee Boiler Corp.
**National Radiator Corp.*
Petroleum Heat & Power Co.
Sarco Co.
**Standard Sanitary Mfg. Co.*
The Thatcher Co.
Thermo Service, Inc.
Titusville Iron Works Co.
**United States Radiator Corp.*
The Vapor Heating Co.
The Vapor Engineering Co.
and many others.

**Vinco Distributors.*

Distributed by Boiler Manufacturers and Jobbers. Sold only in our trade-marked cans like above.

VINCO BOILER CLEANSER	VINCO LEAK SEAL
Packed in 1½, 3, 5 and 10 lb. cans	**Packed in 1 qt. cans only**

VINCO Boiler Leak Seal

A different liquid leak seal. Unique in that it does not induce priming and foaming. It has no unpleasant smell. It makes speedy and permanent repairs of all boiler and heating system leaks. Fine to tighten up new jobs. The directions are simple to follow.

QUANTITIES

Steam and Vapor Systems

Use 1 quart VINCO Liquid Boiler Seal to each 6 square feet *grate* area.

Hot Water Systems

Use 2 quarts VINCO Liquid Boiler Seal to each 6 square feet *grate* area.

What Vinco Does

VINCO permanently removes all the oil, grease, scale and dirt from the internal surfaces and from the boiler water *without the labor of blowing boilers over the top.*

By this thorough cleansing Vinco stops foaming, priming, surging, incomplete circulation and poor radiation.

How Vinco Works

Each minute grain of VINCO powder absorbs several times its own weight of oil, rust and dirt. These larger grains of absorbed impurities then settle and are blown through the bottom, according to directions on each can.

Vinco Specifications for New and Remodeled Steam and Vapor Systems

Cleaning the System—Upon completion of the installation, the contractor shall clean the system by the VINCO method to remove oil, grease, rust and dirt from the boiler using, †...... lb. of VINCO, in exact accordance with manufacturer's directions.

When VINCO is first introduced, have a low fire—just enough for the water to simmer—for three or four hours. After this, pressure can be raised.

This compound must remain in the boiler for 36 actual steaming hours, which corresponds to six or seven days average operation. At the end of this period, boiler must be thoroughly drained and flushed before refilling with clean water.

†In writing specification, insert in this space number of pounds of VINCO to be used in accordance with the following schedule. For systems having:

		*
Up to	350 sq. ft. of ‡radiation	3 lb.
351 "	600 " " "	5 "
601 "	1100 " " "	8 "
1101 "	1400 " " "	10 "
1401 "	1800 " " "	13 "
1801 "	2100 " " "	15 "
2101 "	2700 " " "	18 "
2701 "	3100 " " "	20 "
3101 "	3700 " " "	23 "
3701 "	4200 " " "	26 "
4201 "	4600 " " "	28 "
4601 "	5000 " " "	30 "

Vinco Specifications for Hot Water Systems

Cleaning the System—Upon completion of the installation, the contractor shall clean the system by the VINCO method using **........lb. of VINCO in exact accordance with manufacturer's special directions for hot water systems, given on their cans.

**Only one-half quantities in specification table required for hot water systems.

Vinco for Old Systems

Annual cleaning of the old heating system adds years of life to the boiler, prevents rust deterioration and saves much fuel and fire attendance.

Our Free Laboratory Service

Saves thousands of dollars by analyzing the boiler water *before* making needless mechanical changes. If desired, the boiler water is again examined *after* the VINCO treatment is completed, and "certified chemically correct" for boiler operation. Then if the boiler still primes, foams, or surges, look for mechanical flaws. (Write for details.)

Our Three-fold Guarantee

1. VINCO contains no potash, lye, soda of any kind, oil, acid, or other harmful ingredients. (*See most recent publications of leading boiler manufacturers advising against use cf acids or alkali in boilers*).
2. VINCO meets every performance claim. Purchase price is refunded if results are not entirely satisfactory when VINCO has been used according to directions.
3. Your time, money and comfort are further safeguarded by our free laboratory service.

Above 5000 sq. ft. use an additional pound of VINCO for each additional 300 sq. ft. of radiation.

‡In determining amount of VINCO to be used all radiation may be taken at actual rating.

**For old systems use only one-half quantities given in specification table.*

M^c^DONNELL & MILLER

Manufacturers of

McDONNELL ***Boiler Water Level*** CONTROL

Wrigley Building, **Chicago**

"Doing one thing well"

Automatic Boiler Water-Level Control

McDonnell Boiler Water-Level Controls protect low pressure boilers from the costly repairs and shutdowns that result when someone forgets the boiler water line, and end the trips up and down the basement stairs to watch the water glass.

In all of the controls illustrated, the feed valve and working parts are removed from the heat of the float chamber—all are packless and have stainless steel valves. These features are behind the acknowledged dependability of McDonnell Water-Level Controls.

McDonnell Safety Feeders for Hand-Fired Boilers

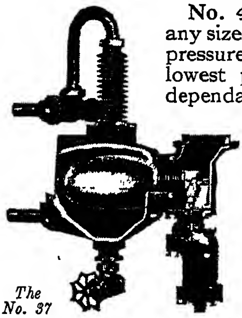
No. 37—Safety Feeder—For boilers up to 2,500 sq. ft., maximum steam pressure, 15 lbs. Is installed in gauge-glass tappings—no hack saw needed. Cuts installation time from hours to minutes.

No. 30-S—Safety Feeder—For boilers from 2,500 to 5,000 sq. ft., maximum steam pressure, 15 lbs. The ideal feeder for boilers of this size.

No. 30-L—Safety Feeder—For boilers above 5,000 sq. ft., maximum steam pressure, 25 lbs.

No. 30-H—For all boilers where steam pressure is 25 to 50 lbs. Same design and capacity as No. 30-L, but of heavier construction to stand higher pressure.

Low-Water Cut-Offs

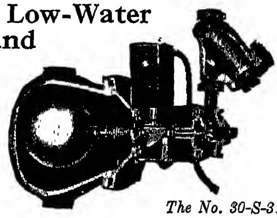


The No. 37

No. 48—For boilers of any size. Maximum steam pressure, 25 lbs. The lowest priced thoroughly dependable low-water cut-off. (110 or 220 volt circuit).

No. 38—Especially recommended for round boilers. Maximum steam pressure, 25 lbs. Has quick hook-up feature.

Combined Low-Water Cut-Offs and Pressure Controls



The No. 30-S-31

No. 36—Switch—A combination low-water cut-off and pressure control (2 to 14 pounds) for use on No. 37 Feeder.

No. 33—Switch—Same as No. 36 except has special mounting plate for use on Nos. 30-S or 30-L.

Combined Feeders and Low-Water Cut-Offs for Automatic Jobs

(These combinations give low water protection with added convenience of automatic water supply).

No. 37-35—For boilers up to 2,500 sq. ft., maximum steam pressure, 15 lbs. The No. 37 Feeder with No. 35 Low-Water Cut-Off Switch attached. (Switch may also be purchased separately and added to any No. 37).

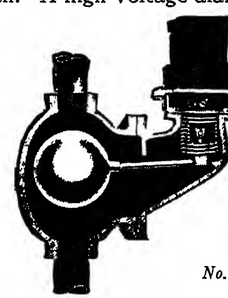
No. 30-S-31—For boilers from 2,500 to 5,000 sq. ft., maximum steam pressure, 15 lbs. The No. 30-S Feeder with switch attached.

No. 30-L-31—For boilers above 5,000 sq. ft., maximum steam pressure, 25 lbs. (No. 31 switch may be purchased separately and added to any Nos. 30-S or 30-L Feeder).

Low-Water Alarms

No. 40—Switch. A high voltage alarm switch. Can be furnished with mountings for use on Nos. 37, 30-S, 30-L, 30-H, or 38.

No. 41—Combination low-water cut-off and low voltage alarm switch for use on Nos. 37, 30-S, 30-L, 30-H, or 38.



No. 48

Our Engineering Department offers prompt advisory service. Descriptive literature, engineering data, and installation drawings upon request

Burnham Boiler Corporation

Irvington-on-Hudson, New York

New York Office: Graybar Building

Offices: Boston; Philadelphia; Chicago; Queens Village, L. I.; San Francisco; Baltimore; Springfield; Lancaster; Pittsburgh; Zanesville; Elizabeth

Plants at Elizabeth, N. J.; Lancaster, Pa.; Zanesville, Ohio

There's a Burnham for every Heating purpose

1—Water Tube Boilers for Steam and Hot Water Heating.

17, 21, 27 and 36 in. Double shaking grates and long fire travel. Rating to 9,050 sq. ft. for steam and 15,085 sq. ft. for water.

2—Water Tube Boilers Jacketed in Color.

17, 21 and 27 in. Steel Jacket and 4-ply air cell asbestos insulation. Enameled rich red. Jacket goes on after all other set up work. Rating to 4,225 sq. ft. for steam and 6,800 sq. ft. for water.

3—Big Twin Sectional Boilers.

50 in. Grate, divided for easy shaking. Twin sections, divided down the middle. Ratings to 19,450 sq. ft. for steam, 31,800 sq. ft. for water.

4—Tube Type Smokeless Boilers.

For burning soft coal efficiently and without smoke. Meet smoke ordinances everywhere. Similar to (1) above, with addition of smokeless feature.

5—Welded Steel Boilers—Also Three-Purpose Welded Steel Boilers.

For heating, hot water supply and incineration. Coal or oil. Completely welded for 15 lbs. working pressure. Multiple shaking grates. Sizes for commercial or domestic uses. Special folder sent on request.

6—Round Sectional Boilers.

This boiler made the long fire travel famous. Handled easily. Very large steam dome. Ratings up to 1,550 sq. ft. for steam, 2,560 for water.

7—High Pressure Hot Water Supply Boilers.

Sectional construction Guaranteed to 80 lbs. working pressure. Supplies up to 3,800 gallons.

8—Junior Hot Water Supply Boilers.

Will keep 175 to 700 gallon tank always full of hot water. Guaranteed to 80 lbs. working pressure.

9—Burnham-Simplex Gas Boiler.

With the patented flue ways, tapering section fins, air control and cast-iron draft diverter. Highly efficient. Steam boiler A.G.A. ratings from 290 to 5250 sq. ft. Water boiler A.G.A. ratings from 470 to 8400 sq. ft.

10—Burnham Oil-Burning Boilers.

A specific-sized boiler for each specific heat job for use with any standard oil burner. Round Sectional Burnhams in 6 Series and 24 sizes. Square Burnhams in 5 Series and 39 sizes. For steam, vapor or water.

11—Burnham-Taco Tanks.

Combining water heater and storage tank in one unit for summer-winter use. Removable copper heating element. Tanks may be galvanized, everdur or copper.

12—Burnham Cast-Iron Radiators.

Occupy about ⅓ less space than ordinary cast-iron radiators of same rating. Shorter. Lower. Narrower. 3-tube type, 3¼ in. wide. 4-tube type 4$\frac{7}{16}$ in. wide.

13—Fero Tube Radiators.

All heights—3, 4, 5, 6 and 7 tubes.

14—Burnham Air and Vacuum Valves.

Full line for radiators, risers and mains.

15—Complete Line of Heating Accessories.

Including steel tanks of all kinds.

Catalogs Sent on Request

AMERICAN RADIATOR COMPANY

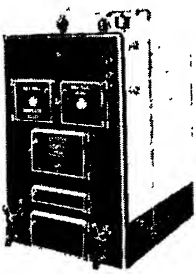
40 West 40th Street, New York, N. Y.

Division of AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

AMERICAN RADIATOR PRODUCTS

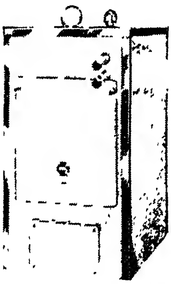
BOILERS

IDEAL REDFLASH BOILERS



Completely equipped, sectional and enclosed in red enameled steel jacket, the Ideal Redflash Boiler is made in a range of sizes to fit homes and other buildings, large or small. It combines the latest scientific improvements with economy, cleanliness and attractiveness.

IDEAL OIL BURNING BOILER

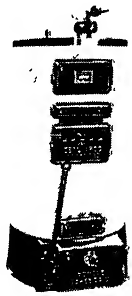


With its attractive green enameled steel jacket, trimmings and fittings in glistening chromard finish, the Ideal Oil Burning Boiler No. 12 brings new beauty and efficiency to oil burning. Hot gases must travel four times the length of the boiler.

IDEAL HOT-WATER SUPPLY

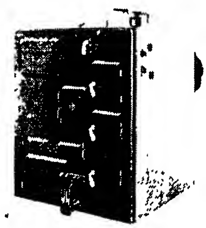
Ideal Hot-Water Supply Boilers are designed for use where large quantities of hot-water must be constantly available. They are made of cast-iron and are virtually immune to rust and corrosion. They burn all types of fuel.

NEW IDEAL ARCO ROUND



Retaining all fundamental features which made its predecessor the standard round boiler of America, the New Ideal Arco Round now comes completely equipped and enclosed in a stippled red enameled jacket. A new device, the Arco Circulator, sets up positive water circulation within the boiler.

IDEAL MAGAZINE BOILERS



Gravity-feeding and automatically regulated, Ideal Magazine Boilers, Nos. 15 and 25, burn coke, anthracite or a mixture of either fuel with buckwheat or pea coal. They will run 12 to 24 hours on one fuel charge, depending upon outside temperature.

IDEAL WATER TUBE

Because of its sectional construction, the Ideal Water Tube Boiler can be installed in old buildings without difficulty. An extensive series of water-backed, vertical tubes expose an unusually large amount of heating surface.

SEE IDEAL FITTER for Complete Information on American Radiator Products

AMERICAN RADIATOR COMPANY

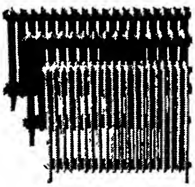
40 West 40th Street, New York, N. Y.

Division of AMERICAN RADIATOR & STANDARD SANITARY CORPORATION

AMERICAN RADIATOR PRODUCTS

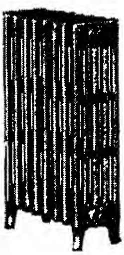
RADIATORS

ARCO RADIATORS



The New Arco Radiator is equally adaptable for exposed, recessed or concealed installations. It occupies approximately one-third less space than older models, yet it provides the same heating output. Its slender, graceful lines and compact, sturdy construction contribute another step in modern heating design.

CORTO RADIATORS



Made of cast-iron, Corto Radiators are durable and time tested. Their threaded nipple construction is known for tightness and permanence. The scientific internal design assures unhampered passage of steam or water. And a severe hydraulic pressure test guarantees every Corto free from defect.

FANTOM RADIATORS

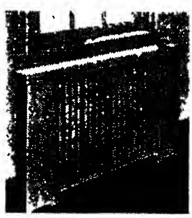
Blending with the wall, the Fantom is a cast-iron radiator to be hung on brackets under a window. Recessed, partly recessed or fully exposed, it radiates heat from its exposed surface and also sends upward a curtain of warm air which blankets off window drafts.

ARCO CONVECTOR

The Arco Convector is a cast-iron radiator for complete concealment. Ample inner space assures positive air elimination and unhampered flow of steam or water. Fins are scientifically proportioned and spaced to allow proper air flow. A full line of Arco Enclosures are designed for the Arco Convector.

ARCO ENCLOSURES

Arco Radiator Enclosures are designed either entirely to conceal radiators such as New Murray, Arco and Arco Convector; or to frame such radiators as the Arco, Corto or Fantom. In each instance Arco Enclosures give a "tailor-made job." All enclosures give easy access for cleaning and regulation.



Type "OK" Enclosures are made in a complete range of sizes to fit Arco, Corto and Peerless Radiators of all dimensions. No cutting or fitting required. They need only be ordered in proper length, width and height and quickly set in place.

SEE IDEAL FITTER for Complete Information on American Radiator Products

CRANE CO.

Manufacturers of Valves, Fittings, Fabricated Piping, Steam Specialties, Plumbing and Heating Materials

836 S. Michigan Avenue **Chicago, Ill.**

Branches in All Principal Cities

Write for catalogues and full information about any materials in which you are interested

Crane Boilers

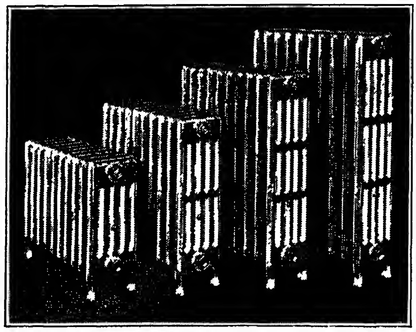
Styles—Crane boilers are made in round and sectional styles . . . . in capacities to efficiently and economically meet the heating requirements of every type of structure.

Complete Line—Crane supplies everything for the entire heating system—boilers, radiators, valves, fittings, piping, and specialties. Uniformity in quality and efficiency is assured.

Economical Firing—Elongated two pass gas flues are arranged for controlled water travel and have 50 per cent more ceiling heating surface. Baffles direct water across top of combustion chamber controlling as well as lengthening water travel and keeping walls scrubbed free of insulating film. Grate areas, combustion chambers, and heating surfaces are proportioned for longest firing periods.

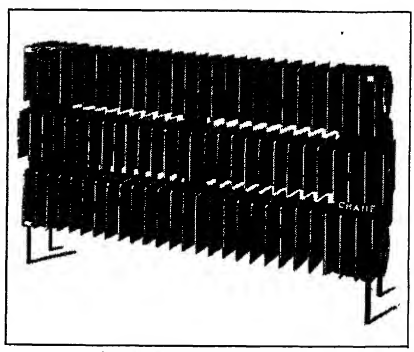
Easy Cleaning—Broad, flat gas flues have less tendency to collect soot and permit easy entrance for scraper. Soot drops into the first pass when scraped and from there it can be pushed into the combustion chamber.

Sectional View of Square Boiler



Group of Radiators

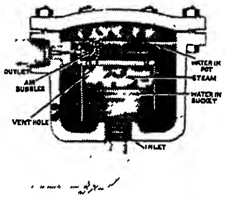
Crane radiation is made in styles and sizes to meet all building requirements. The Crane radiator has established a reputation for high heating efficiency, durability, ease of assembly, and decorative grace. The addition of the Crane Invisible Shield places this radiator in a class by itself. This shield directs the warmed air into the living zone instead of to ceiling and, consequently, reduces fuel consumption.



Concealed Radiator with legs

Crane Concealed Radiators are exceptionally compact and made in sizes to suit recess requirements. Cast in one piece of cast-iron. Unaffected by expansion or contraction.

No. 981 Inverted Open Float Steam Traps



Sectional View No. 981 Trap

This is a new line of very efficient steam traps of large capacity yet low in price for draining condensation from various steam heated apparatus such as garment presses, coffee urns, jacketed kettles, unit heaters, laundry machinery, heating coils, etc. Under the most exhaustive service tests, these traps have proven themselves dependable and economical in first cost, installation and maintenance.

Operation is as follows:

Condensation flows upward into the trap until the body is full. Then it is automatically discharged through a valve at the top. When no more condensation enters, steam displaces the water in the float, which then becomes buoyant and rises, thus closing the valve and stopping the discharge until an accumulation of condensation enters the float and causes it to sink.

Capacities in Pounds of Water per Hour

Pressure at Inlet	½ In. Size	¾ In. Size	1 In. Size
5	915	3050	12300
10	880	3500	14000
15	1080	4300	17500
30	960	3800	15300
50	680	2700	10900
75	350	2250	9060
100	405	2650	10450
125	450	1850	7370
150	460	2050	8000

Crane Welding Fittings

90° Ell

Crane Co. manufactures a complete line of welding fittings and flanges in all sizes from 1 in. to 18 in. inclusive. Elbows and Return Bends are made of seamless steel tubing in standard and extra strong pipe thickness. Tangents are provided at each end for ease of lining up by the use of ordinary jigs and clamps and to bring the weld away from the point of curvature.

Tee

45° Ell

Tees are forged seamless in sizes 2½ in. and smaller. Sizes 3 in. to 10 in. are forged in halves and electrically welded.

Forged Steel Welding Neck Flange

Welding neck flanges may be had in various weights and with various facings to suit all conditions of service.

Forged Steel Slip-on Flange

Return Bend

Cranelap Welding Nipples, which are short lengths of seamless steel pipe provided with a full thickness Cranelap on one end and beveled for welding on the other end, are made of standard and extra strong pipe. Proper weight forged steel Cranelap flanges to suit the required service can be furnished with these nipples.

Cranelap Welding Nipple and Cranelap Flange

In addition to the fittings listed, welding caps, reducing nipples and reinforcing saddle flanges are available in a wide range of sizes.

National Radiator Corporation

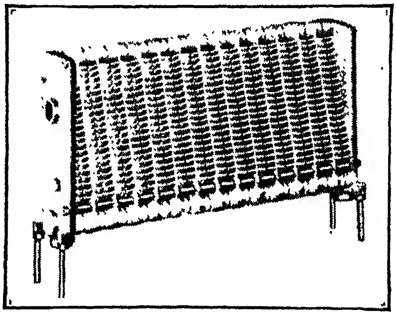
MANUFACTURERS OF BOILERS, RADIATORS AND CONVECTORS

Johnstown, Pa.

National Service through these Branch Offices and Warehouses:

Baltimore, Md.	2622 Matthews St.	Milwaukee, Wis.	511 E. Mason St.
Boston, Mass.	250 Stuart St.	New York, N. Y.	55 West 42nd St.
Buffalo, N. Y.	904 Main St.	Philadelphia, Pa.	2124 Arch St.
Chicago, Ill.	1111 East 83rd St.	Pittsburgh, Pa.	339 Blvd. of the Allies
Cincinnati, Ohio	3530 Spring Grove Ave.	Richmond, Va.	3032 Norfolk St.
Cleveland, Ohio	4500 Euclid Ave.	Washington, D. C.	2205 Fifth St. N.E.

AERO CAST IRON CONVECTORS



The Convector with a Guaranteed Rating

Tested and Rated in Accordance with the A.S.H.V.E. Standard Code for Testing and Rating Concealed Gravity Type Radiation (Steam Code).

U. S. Patents No. 1,888,545 No. 1,912,556. Other Patents Pending

A message to Heating and Ventilating Engineers, Heating Contractors and Architects

Aero Convector Ratings

Aero Convectors are rated in accordance with the American Society of Heating and Ventilating Engineers' Standard Code for Testing and Rating Concealed Gravity Type Radiation.

These Ratings are based on the Condensation from the convector, the Latent Heat in the steam and the Correction Factor as provided in the Code.

The Catalog Ratings of Aero Convectors are Guaranteed when:

1. The Convectors are used with enclosures having dimensions as shown in the current catalog.
2. The Convectors are placed at a height in the enclosures corresponding to that obtained with "low legs."
3. The Convectors are completely filled with steam at a temperature of 215°F. or water at a temperature of 175°F.

Other Methods of Rating

Present day methods of rating concealed heating units are not uniform, due to the addition to actual condensation ratings of an intangible called "Heating Effect" on the theory that less Equivalent Direct Radiation is required with a concealed radiator than with an exposed radiator.

The Code makes no provision for determining ratings on the basis of "Heating Effect," "Available Heat" or "Effective Heat."

Results of a State University Investigation of Heating Effect

Recent tests conducted at a State University developed these two conclusions which disprove the "Heating Effect" theory:

1. The steam condensation obtained when the temperature is maintained at 68°F, at the 30 inch level is an approximate measure of the relative effectiveness of different types of heating units in providing human comfort; lower condensation corresponding to greater effectiveness.
2. The heating effect is not materially greater than the total heat output as measured by steam condensation under given standard conditions, and the practice of adding a large proportion to condensation rating in order to provide for heating effect CANNOT BE JUSTIFIED.

For Your Protection

When specifying convectors be sure of the basis of rating. Insist that Ratings be according to the A.S.H.V.E. Standard Code. Be safe and choose Aero Convectors with catalog ratings that are guaranteed.

Write or 'phone nearest Branch Office for copy of Aero Convector Catalog No. 2 and Supplement "A"

National Radiator Corporation

General Offices, **Johnstown, Pa.**

NATIONAL PREMIER STEEL BOILERS

Constructed for 15 lbs. working pressure in accordance with the A. S. M. E. Code

SMALL STEEL BOILERS—REAR SMOKE OUTLET—THREE PASS

Type "DB" Solid Fuel—Hand Fired; Type "MB" Solid Fuel—Stoker Fired; Type "OB" Oil or Gas Fired

Boiler No. —Types "DB," "MB," "OB"	251	252	253	254	255	256	311	312	313	314	315	316
Steam Rating—Type "DB" ....Sq. Ft.	485	645	840	990	1120	1250	1350	1525	1700	1875	2050	2225
Steam Rating—Types "MB," "OB"..Sq. Ft.	735	890	1045	1200	1355	1510	1635	1850	2065	2280	2495	2710
Water Rating—Type "DB" ...... .Sq. Ft.	775	1030	1340	1580	1790	2000	2160	2440	2720	3000	3280	3560
Water Rating—Types "MB," "OB"..Sq. Ft.	1175	1425	1675	1925	2175	2425	2615	2960	3305	3650	3995	4340
Stack Diameter ....In.	12	12	12	12	12	12	16	16	16	16	16	16
Stack Height .... .Ft.	35	35	35	40	40	40	40	40	40	45	45	45

REAR SMOKE OUTLET—THREE PASS

Type "DB" (Direct Draft); Type "SB" (Smokeless); Type "OB" (Oil, Gas or Stoker Fired)

Boiler No. Types "DB," "SB" and "OB"	S.H.B.I. Steam Ratings Type "DB," and "SB"	S.H.B.I. Steam Ratings Type "OB"	Heating Surface	Grate Area Types "DB" and "SB"	Furnace Volume Above Water Leg Ring Type "OB"	Gross Base Volume Floor to Water Leg Ring Type "OB"	Height of Water Line	Width of Boiler	Length of Boiler	Length Overall	Size of Steam Outlet	Size of Return	Height of Rear Smoke Outlet	Height Overall	Dia. of Rear Smoke Outlet	Dia. of Breeching, 1 Boiler	Dia. of Stack, 1 Boiler	Height of Stack, 1 Boiler
	Sq. Ft.	Sq. Ft.	Sq. Ft.	Sq. Ft.	Cu. Ft.	Cu. Ft.	Ft. In.	In.	Ft. In.	Ft. In.	In.	In.	Ft. In.	Ft. In.	In.	In.	In.	Ft.
371	2550	3100	183	11.6	28.3	16.5	5–8½	37	5–4	6–4	6	4	5–4	6–7	22	22	20	50
372	3000	3675	216	11.6	31.8	19.1	5–8½	37	6–1	7–1	6	4	5–4	6–7	22	22	20	50
373	3500	4250	250	11.6	36.1	22.0	5–8½	37	7–0	8–0	6	4	5–4	6–7	22	22	20	55
431	4000	4900	287	12.5	45.2	25.5	5–11	43	7–0	8–0	6	4	5–7	7–0	24	24	22	55
432	4500	5500	323	14.0	49.8	28.2	5–11	43	7–9	8–9	6	4	5–7	7–0	24	24	22	55
433	5050	6125	360	15.5	54.3	30.9	5–11	43	8–6	9–6	6	4	5–7	7–0	24	24	22	60
491	6000	7300	428	16.3	58.7	29.0	6–9	49	7–0	8–2	8	5	6–5	8–2	26	26	24	60
492	6550	7950	467	18.0	63.4	31.4	6–9	49	7–7	8–9	8	5	6–5	8–2	26	26	24	60
493	7000	8500	500	19.8	66.4	33.5	6–9	49	8–1	9–3	8	5	6–5	8–2	26	26	26	65
551	8500	10350	608	20.6	83.5	48.4	7–1	55	8–4	9–6	8	5	6–10	8–6	28	23	26	65
552	9575	11625	684	22.6	92.0	53.7	7–1	55	9–3	10–5	8	5	6–10	8–6	23	23	28	65
611	10840	13150	774	25.5	97.8	52.5	7–11	61	8–2	9–5	8	6	7–4	9–4	32	32	30	70
612	12000	14575	857	25.5	106.6	57.3	7–11	61	8–11	10–2	8	6	7–4	9–4	32	32	30	70
671	14330	17400	1024	28.3	131.6	68.9	8–0	67	9–9	11–1	8	6	7–7	9–6	34	34	32	70
672	16350	19850	1168	30.8	145.7	77.1	8–0	67	10–11	12–3	8	6	7–7	9–6	34	34	32	70
731	18650	22650	1332	33.9	172.0	85.9	8–3½	73	11–2	12–10	8	6	7–9	9–10	36	36	34	80
732	20700	25175	1480	36.6	189.3	94.2	8–3½	73	12–3	13–11	8	6	7–9	9–10	36	36	34	90
791	22650	27500	1618	36.9	218.1	103.3	8–5½	79	12–5	14–1	10	6	8–0	10–2	40	40	38	90
792	25550	31000	1825	39.9	241.4	115.1	8–5½	79	13–10	15–6	10	6	8–0	10–2	40	40	38	90
851	28750	34900	2054	40.1	233.3	107.7	9–7½	85	12–0	13–8	10	6	9–0	11–5	42	42	40	100
852	33000	40050	2357	43.3	258.6	121.1	9–7½	85	13–6	15–2	10	6	9–0	11–5	42	42	40	100
853	34150	41450	2438	46.5	266.0	124.8	9–7½	85	13–11	15–7	10	6	9–0	11–5	42	42	40	100

FRONT SMOKE OUTLET—TWO PASS

Type "DF" (Direct Draft); Type "SF" (Smokeless); Type "OF" (Oil, Gas or Stoker Fired)

Boiler No. Types "DF," "SF" and "OF"	S.H.B.I. Steam Ratings Types "DF," and "SF"	S.H.B.I. Steam Ratings Type "OF"	Heating Surface	Grate Area Types "DF," and "SF"	Furnace Volume Above Water Leg Ring Type "OF"	Gross Base Volume Floor to Water Leg Ring Type "OF"	Height of Water Line	Width of Boiler	Length of Boiler	Length Overall	Size of Steam Outlet,	Size of Return	Height of Smoke Outlet	Height Overall	Size of Smoke Outlet	Dia. of Breeching, 1 Boiler	Dia. of Stack, 1 Boiler	Height of Stack, 1 Boiler
	Sq. Ft.	Sq. Ft.	Sq. Ft.	Sq. Ft.	Cu. Ft.	Cu. Ft.	Ft. In.	In.	Ft. In.	Ft. In.	In.	In.	Ft. In.	Ft. In.	In.	In.	In.	Ft.
371	3000	3650	214	11.6	37.9	21.0	5–8	37	5–5	6–9	6	4	5–10	6–7	10x26	20	18	50
372	3500	4250	250	11.6	41.8	23.6	5–8	37	6–3	7–7	6	4	5–10	6–7	10x26	20	18	55
373	4000	4900	286	12.9	45.8	26.2	5–8	37	7–1	8–5	6	4	5–10	6–7	10x26	20	18	55
431	4500	5500	322	14.0	61.1	32.2	5–10½	43	7–6	8–11	6	4	6–0½	7–0	11x30	22	20	60
432	5000	6100	358	15.5	65.1	34.6	5–10½	43	8–2	9–7	6	4	6–0½	7–0	11x30	22	20	60
433	6000	7300	429	16.9	73.6	39.7	5–10½	43	9–7	11–0	6	4	6–0½	7–0	11x30	24	22	60
491	7000	8500	500	18.1	80.1	36.6	6–9	49	7–4	9–0	8	5	6–11	8–2	14x36	26	24	65
492	8500	10300	607	21.3	91.2	42.5	6–9	49	8–9	10–5	8	5	6–11	8–2	14x36	26	24	65
551	10000	12150	715	22.6	114.3	61.4	7–1	55	9–0	10–9	8	5	7–3	8–6	15x40	28	26	70
552	12500	15200	893	26.6	132.8	73.0	7–1	55	11–0	12–9	8	5	7–3	8–6	15x40	30	28	75
611	15000	18200	1072	28.0	155.4	77.0	7–11	61	10–3	12–2	8	6	8–1	9–4	17x42	32	30	80
671	17500	21250	1250	30.9	181.6	91.4	8–1	67	11–2	13–1	8	6	8–3	9–6	17x46	34	32	85
731	20000	24300	1429	33.9	222.4	101.5	8–4	73	11–5	13–4	8	6	8–6	9–10	17x50	36	34	90
791	25000	30400	1786	37.0	278.4	121.6	8–6	79	12–10	14–9	10	6	8–8	10–2	17x56	40	38	90
851	30000	36450	2143	43.3	295.7	124.5	9–7	85	11–10	14–0	10	6	9–9	11–5	20x60	42	40	100
852	35000	42500	2500	46.5	326.6	140.9	9–7	85	13–8	15–10	10	6	9–9	11–5	20x60	42	40	100

Ratings conform with the Steel Heating Boiler Institute's Code for Rating Low Pressure Heating Boilers.
S. H. B. I. Water Ratings are 60 per cent greater than S. H. B. I. Steam Ratings.

Spencer Heater Company

Division of Cord Corporation

Williamsport, Pa.

New York, N. Y.	Albany, N. Y.	Philadelphia, Pa.	Milwaukee, Wis.
Buffalo, N. Y.	Syracuse, N. Y.	Bethlehem, Pa.	Cincinnati, Ohio
Chicago, Ill.	Boston, Mass.	Scranton, Pa.	Birmingham, Ala.

L-2 Series Spencer Cast-Iron Sectional Boiler

Minneapolis No. 40 Thermostat and electric damper motor furnished as standard equipment with all Spencer Cast-Iron Boilers.

Spencer Magazine Feed Boilers

The original magazine feed boiler with sloping grates, a gravity stoker boiler. The water jacketed magazine holds enough fuel for 12 to 24 hours. Fuel feeds automatically by gravity as fast or as slow as the fire requires. Uniform depth of fire bed keeps fire always at most efficient combustion point, giving maximum efficiency at minimum fuel cost.

Spencer Boilers are made in sizes to suit every home or building, large or small. The direct cast-iron column radiation load which each size Spencer will carry is guaranteed.

Automatic Heat at Lowest Cost

Spencer Boilers are designed especially to burn small-size, low-cost fuels, such as No. 1 Buckwheat anthracite or small size by-product coke. No. 1 Buckwheat anthracite, for example, costs as much as $4.00 less a ton than the larger sizes.

Spencer Rotary Ash Receiver

affords an ideal method of ash disposal. Ashes are raked directly from the ash pit into cans, contained in a water-tight steel tank sunk in the basement floor. Eliminates dust and ashes about the cellar.

Spencer Combination Boiler

designed to burn coal, coke or gas with equal efficiency. Possible to change instantly from coal or coke to gas; specially designed gas burner is always there, ready for instant use, nothing to attach or connect. Furnished complete with burner, Minneapolis No. 77 8-day room temperature thermostat, pressuretrol, vaporstat or aquastat, throttling valve and automatic pilot control.

Spencer Combination Boiler for Coal, Coke or Gas

GUARANTEED CAPACITIES AND DIMENSIONS

Type	Boiler No.	Direct Cast-Iron Column Radiation Loads, Sq. Ft.* Steam	Water	Tank Capacity Gallons	Grate Area	Outlets	Returns	Chimney Flue	Diameter Smoke Pipe	Over-all Dimensions Length	Width	Height
J Type	J-3	175	290	240	1.30	1-3″	2-3″	8″x 8″x35′	8″	28½″	24″	45″
	J-4	265	440	390	1.90	1-3″	2-3″	8″x 8″x35′	8″	35½″	24″	45″
	J-5	355	590	540	2.50	1-3″	2-3″	8″x 8″x35′	8″	42½″	24″	45″
L-1 Type	L-105	390	645	600	2.60	1-4″	2-4″	8″x 8″x35′	10″	40½″	32″	57″
	L-106	510	845	810	3.33	2-4″	2-4″	8″x 8″x35′	10″	47½″	32″	57″
	L-107	630	1,045	1,020	4.07	2-4″	2-4″	8″x12″x35′	10″	54½″	32″	57″
L-2 Type	L-205	550	910	900	3.63	1-4″	2-4″	8″x12″x35′	10″	40½″	40″	60″
	L-206	725	1,200	1,170	4.66	2-4″	2-4″	8″x12″x35′	10″	47½″	40″	60″
	L-207	900	1,490	1,440	5.68	2-4″	2-4″	8″x12″x40′	10″	54½″	40″	60″
	L-208	1,075	1,780	1,740	6.70	2-4″	2-4″	12″x12″x40′	10″	61½″	40″	60″
	L-209	1,250	2,070	2,040	7.73	2-4″	2-4″	12″x12″x40′	10″	68½″	40″	60″
L-3 Type	L-305	1,150	1,900	.....	7.29	2-4″	2-4″	12″x12″x40′	14″	42″	56″	65″
	L-306	1,500	2,475	.....	9.33	2-4″	2-4″	12″x12″x40′	14″	49″	56″	65″
	L-307	1,850	3,050	.....	11.37	2-4″	2-4″	12″x12″x40′	14″	56″	56″	65″
	L-308	2,200	3,625	.....	13.41	2-4″	2-4″	12″x16″x45′	14″	63″	56″	65″
	L-309	2,550	4,200	.....	15.45	2-4″	2-4″	12″x16″x45′	14″	70″	56″	65″
	L-310	2,900	4,775	.....	17.49	2-4″	2-4″	16″x16″x50′	14″	77″	56″	65″
	L-311	3,250	5,350	.....	19.53	2-4″	2-4″	16″x16″x50′	14″	84″	56″	65″

*This includes ample provision for heat loss in covered mains, risers and returns, and for peak loads, as covered by guarantee.

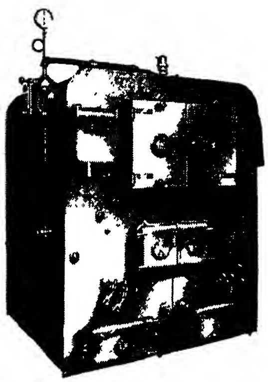
Spencer Heavy Duty Steel Tank Heater

Spencer Heavy Duty Tank Heaters

Furnished in both cast-iron sectional and steel tubular types. Cast-iron sectional built with extra heavy sections, and steel tubular with copper-bearing steel plates and Toncan Iron tubes. For working pressure up to 120 pounds.

Spencer Steel Tubular Boilers

are combination water and fire tube construction, built of heavy steel plates and copper bearing tubes. Improved and perfected through more than thirty-five years successful operation.



M-7 Series Spencer Steel Tubular Boiler

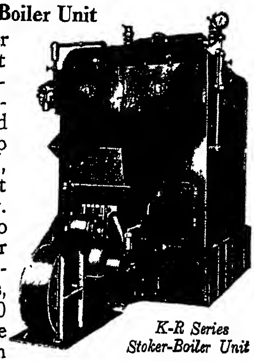
K-L Series Oil Burning Boiler

Spencer Oil Burning Boiler

Specially designed to meet every requirement for efficient oil burning. Boiler, combination water tube and fire tube; welded steel construction; built in two sections for easier installation and handling. Capacities, 2250 to 40,000 sq. ft. Ask for detailed data.

Spencer Stoker-Boiler Unit

Consists of boiler and stoker, built as a unit, automatically controlled. Designed to burn cheap grades of coal, smokelessly, at high efficiency. Requires no brick work or concrete in setting. Capacities, 2250 to 40,000 sq. ft. Literature and data sent on request.



K-R Series Stoker-Boiler Unit

Spencer Low Cost Automatic Heating Equipment for Every Fuel and Every Building. Write for catalog.

GUARANTEED CAPACITIES AND DIMENSIONS

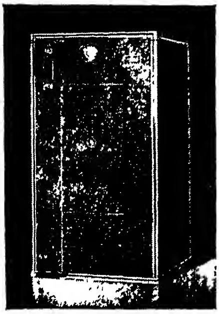
Series	Boiler No.	Direct Cast-Iron Column Radiation Loads, Sq. Ft.* Steam	Tank Capacity Gallons	Grate Area	Outlets	Returns	Chimney Flue	Diameter Smoke Pipe	Overall Dimensions Length	Width	Height
(M4 Series)	M4-2	250	430	2.25	1-3″	1-3″	8″x 8″x35′	10″	32½″	27″	55″
	M4-3	410	710	3.17	1-3″	1-3″	8″x 8″x35′	10″	39½″	27″	55″
	M4-4	570	990	4.09	1-3″	1-3″	8″x12″x35′	10″	46½″	27″	55″
	M4-5	730	1,270	5.02	2-3″	2-3″	8″x12″x35′	10″	53½″	27″	55″
	M4-6	890	1,550	5.94	2-3″	2-3″	8″x12″x35′	10″	60½″	27″	55″
	M4-7	1,050	1,830	6.86	2-3″	2-3″	8″x12″x35′	10″	67½″	27″	55″
(M5 Series)	M5-4	1,200	2,100	8.	1-4″	2-3″	12″x12″x35′	12″	50″	48″	60″
	M5-5	1,400	2,450	9.75	1-4″	2-3″	12″x12″x35′	12″	57″	48″	60″
	M5-6	1,600	2,800	11.50	1-4″	2-3″	12″x12″x40′	12″	64″	48″	60″
	M5-7	1,800	3,150	13.25	1-4″	2-3″	12″x12″x40′	12″	71″	48″	60″
	M5-8	2,000	3,500	15.	1-4″	2-3″	12″x12″x40′	12″	78″	48″	60″
(M6 Series)	M6-6	2,300	.....	14.0	1-6″	2-2½″	12″x12″x40′	14″	65½″	58″	66″
	M6-7	2,600	.....	15.8	1-6″	2-2½″	12″x16″x45′	14″	72½″	58″	66″
	M6-8	2,900	.....	17.6	1-6″	2-2½″	12″x16″x45′	14″	79½″	58″	66″
	M6-9	3,200	.....	19.4	1-6″	2-2½″	16″x16″x50′	14″	86½″	58″	66″
	M6-10	3,500	.....	21.2	1-6″	2-2½″	16″x16″x50′	14″	93½″	58″	66″
(M7 Series)	M7-6	4,000	.....	19.0	1-8″	2-3″	20″x20″x60′	18″	75½″	68″	80″
	M7-7	4,700	.....	22.0	1-8″	2-3″	20″x20″x60′	18″	82½″	68″	80″
	M7-8	5,400	.....	25.0	1-8″	2-3″	20″x20″x65′	18″	89½″	68″	80″
	M7-9	6,100	.....	28.0	1-8″	2-3″	22″x22″x70′	18″	96½″	68″	80″
	M7-10	6,800	.....	31.0	1-8″	2-3″	22″x22″x70′	18″	103½″	68″	80″
(M8 Series)	M8-6	8,000	.....	32.36	1-8	2-4″	24″x24″x65′	24″	79¼″	108″	85″
	M8-7	9,750	.....	37.02	1-8″	2-4″	24″x24″x65′	24″	86¼″	108″	85″
	M8-8	11,500	.....	41.68	1-8″	2-4″	30″x30″x70′	24″	93¼″	108″	85″
	M8-9	13,250	.....	46.34	1-8″	2-4″	36″x36″x70′	24″	100¼″	108″	85″
	M8-10	15,000	.....	51.00	1-8″	2-4″	36″x36″x70′	24″	107¼″	108″	85″

*This includes ample provision for heat loss in covered mains, risers and returns, and for peak loads, as covered by guarantee.

UNITED STATES RADIATOR CORPORATION

Manufacturers of Capitol Boilers and Radiators
THE PACIFIC STEEL BOILER CORPORATION
Manufacturers of Pacific Steel Heating Boilers
GENERAL OFFICES: DETROIT, MICHIGAN

The Capitol Oil Burning Boiler—OB Series



RATINGS

STEAM		WATER	
OB-20		OB-20	
Capacity Sq. Ft.	Direct Cast-Iron Radiator Load, Sq. Ft.	Capacity Sq. Ft.	Direct Cast-Iron Radiator Load, Sq. Ft.
825	500	1320	800
OB-25		OB-25	
1190	700	1905	1120

Steam boilers include Capitol Low Water Cut-off, manufactured by the McDonnell-Miller Company and built-in Taco Domestic Hot Water Heater.

Automatic Water Feeder in combination with Low Water Cut-off, extra. Water boilers include built-in Taco Domestic Hot Water Heater.

Capitol Oil Burning Boilers, OB Series, when connected to the direct cast-iron radiator load shown in table, have sufficient reserve capacity to provide for heat loss from piping amounting to 25 per cent of the standing radiation, with an additional 25 per cent reserve for pick-up load, plus the reserve required to raise the temperature of the domestic hot water from 50 degrees to 150 degrees Fahrenheit in three hours. Storage tank capacity has been estimated not to exceed 40 gallons with the OB-20 Boiler and 40 to 85 gallons with the OB-25 Boiler.

Where the domestic hot water supply is not heated by the boiler, the OB-20 Boiler may be connected to 530 sq. ft. of steam radiation, or 850 sq. ft. of water radiation, and the OB-25 may be connected to 763 sq. ft. of steam radiation or 1220 sq. ft. of hot water radiation.

Construction

Made of cast-iron. Heavy Metal Jacket.
Rock Wool Insulation.

Heating Surface

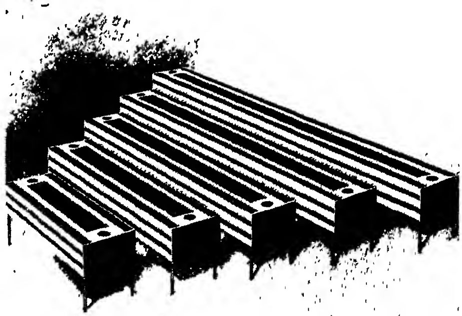
Large combustion space.
Concentrated prime and ribbed heating surfaces.
Seven direction changes of gases—no stratification.
Controlled gas travel—intimate contact of all gases with heating surface.

Operation

Year 'round hot water with built-in Taco.
Low water cut-off.
Enclosed steam trimmings.
Visual fire inspection.
Provision for stack switch and enclosed aquastat.
Sound insulated.

Capitol Fincast Radiators and Enclosures

Made entirely of cast-iron.
Made without joints.
Cast in one piece.
Many lengths and widths.
Tappings—top, bottom or ends.
Complete choice of enclosures.



Williams Oil-O-Matic Heating Corporation

Manufacturers of Automatic and Manually Controlled Fuel Oil Burners

WILLIAMS OIL-O-MATIC HEATING
BLOOMINGTON, ILL.

Bloomington, Ill.

Service to Architects and Builders

CHICAGO, 185 North Michigan Avenue

WILLIAMS OIL-O-MATIC HEATING
NEW YORK, Graybar Building

Williams Oil-O-Matic offers a complete line of oil heating equipment—covering all heating and hot water needs. Also boiler-burner units, heavy duty range burners and unit heaters. Latter used with water heaters where both hot water and heat are required.

SPECIFICATIONS

Model No.	Length, In.	Width, In.	Height, In.	HP.	R.P.M.	Gals. Fuel Oil per Operating Hr. Min.	Max.
Model K-1.5	25	15	22	1/8	1750	½	1½
Model KB (1200)	29	18	20	1/6	1150	1½	3
Model KB (1800)	29	18	20	1/4	1750	2½	5
Model J (1200)	45	20	25	1/4	1150	4	10
Model J (1800)	45	23	25	1/2	1750	8	15
Model JJ	50	33	22	1	1750	12	25
Model K (1200)	34	28	21	1/6	1150	1½	3
Model K (1800)	34	28	21	1/4	1750	2½	5
WATER HEATERS							
WHA*	58	23	28	1/10	1750	..	½
WHB†	75	23	28	1/8	1750	½	1.0
WHC‡	88	29	34	1/8	1750	1	1¾

*Output: 90° rise, 60 gallons per hour. ‡Output: 90° rise, 210 gallons per hour.
†Output: 90° rise, 120 gallons per hour.

In case odd frequency motors are used, the maximum capacity of the burner will be reduced in proportion to the r.p.m. of the motor. The minimum capacity remains the same. Model K must be substituted for KB for odd currents and export

How to Decide Size of Burner

For low pressure domestic boiler, 1 gal. of fuel oil per hour (L40,000 B.t.u.'s) is required for approximately:

Model K-1.5 Oil-O-Matic

300 sq. ft. of steam radiation or its equivalent.

480 sq. ft. of hot water radiation or its equivalent.

70,000 B.t.u.'s when using hot air furnace ratings.

24 sq. ft. steam boiler heating surface (or 2.2 hp.).

1 sq. ft. of grate surface with combustion space 3 ft. high or its equivalent.

For detail data, see Oil-O-Matic Installation and Service Manual.

Domestic Hot Water Supply

Three water heater models cover the domestic and commercial water heating fields. Entire apparatus–including genuine Oil-O-Matic oil burner, combustion chamber, water tank, and all automatic controls —now combined in a single neat and compact unit.

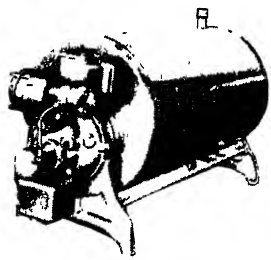
Refer to specifications for dimensions and output capacities of all models—WHA, WHB, and WHC.

Boiler-Burner Units

A complete line of welded steel and also cast-iron sectional Boiler-Burner Units. Wide variety of sizes for steam or hot Water. Write for details.

Underwriters' Listing

Oil-O-Matic burners are listed as standard by National Board of Fire Underwriters to burn oils conforming to A. P. I. standard specifications for Nos. 1, 2, 3, and 4, also Pacific Coast Disel.



Hot Water Supply Unit

Each burner carries Underwriters' label. Also approved by all important codes and governing bodies.

Range Burners

New type Oil-O-Matic range burner, for heavy duty ranges, brings oil heat economy to restaurant, hotel, hospital, steamship, dining car, resort and club.

No priming, no heat generating, no matches, no wicks required; a single lever controls everything.

Engineering Service

Engineering service is available to architects—see A. I. A., File No. 30-GL.

Petroleum Heat & Power Company

Manufacturers of

Petro & Nokol Commercial, Industrial, and Domestic Oil Burner Equipment, Arco-Petro Automatic Boilers for Oil or Gas, and Oil Burner Accessories

Distributors of Fuel Oil

Factory and Main Offices: **Stamford, Conn.**

Branch Offices In

New York, N. Y.
Boston, Mass.
Providence, R. I.
Springfield, Mass.
Newark, N. J.
Philadelphia, Pa.
Baltimore, Md.
Washington, D. C.
Detroit, Mich.
Chicago, Ill.

Subsidiary Companies

Boston Harbor Oil Co.
East Coast Fuel Oil Co.
Power Plant Engineering Co.

OIL BURNERS THAT FIT THE BOILER

A Complete Line of PETRO & NOKOL OIL BURNERS Each Designed for Specific Types of Domestic, Commercial, and Industrial Boilers and Furnaces

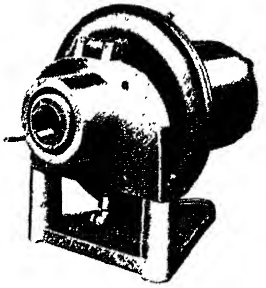
The Petroleum Heat and Power Company recognized early in its 30 years of experience that no one type of burner meets every oil heating requirement in an equally satisfactory way, or solves every oil burner problem.

TYPES OF BURNERS AND RECOMMENDED APPLICATIONS

Model No.	Type	Listed for Oil No.	Gals. Oil per Hr.	Radiation Sq. Ft. Steam	Radiation Sq. Ft. Hot Water	Ignition	Motor H. P.	Price Range and Applications
W-1A W-1B W-1C	Horizontal Direct Motor Driven Rotary Cup Type Burner	3	.5 to 1.0 1.0 to 1.75 1.75 to 2.6	350 615 910	560 985 1450	Gas-Elec. Gas-Elec. Elec.-Gas	1/10 1/10 1/10	New, low cost rotary cup type burner for very small round or square boilers. Also for process steam and hot water service.
K	Wall-wiping Flame Rotary Type Burner	2 (Gas) 2 (Elec.)	.8 to 2.1 .8 to 2.1	285 to 730	460 to 1170	Gas or Elec.	1/20	Minimum cost, fully automatic burner for small round or square boilers and warm air furnaces.
P-1 P-1½ P-2	Pressure Atomizing Gun Type Burner	3 3 3	1.3 to 3.0 2.0 to 5.0 3.0 to 7.5	1050 1750 2620	1680 2800 4190	Continuous Elec.	1/6 1/6 1/4	Low cost, quality built burner for rectangular fire-box, sectional or tubular, boilers or furnaces.
				Radiation Sq. Ft. Steam	Maximum Boiler H. P.			
W-2 W-3 W-4 W-5 W-6 W-7	Horizontal Direct Motor Driven Rotary Cup Type Burner	300 secs. Saybolt Universal at 100°F. max. viscosity or No. 6 Oil preheated	1.5 to 7.0 5.0 to 15.0 10.0 to 25.0 20.0 to 33.0 25.0 to 45.0 25.0 to 62.0	2450 5250 8750 11550 15750 21700 Full Automatic	17½ 37 63 83 113 156 Full Automatic	Elec.-Gas with full automatic Manual with semi-automatic or Manual Control	¼ ½ ½ 1 1½ 2	Manual, semi-automatic, or full automatic heavy duty burner for average Industrial and Commercial installations using heavier grades of oil.
H	Air Turbine Driven Rotary Cup Type Burner	No. 6 Oil preheated	15.0 to 75.0			Manual	1 to 5	Manual and semi-automatic equipment for large multiple burner installations up to any capacity.
M	Mechanical Atomizing Type Burner Natural and Forced Draft	No. 6 Oil preheated	Up to 125.0			Manual		Manual equipment for large multiple burner installations up to any capacity.

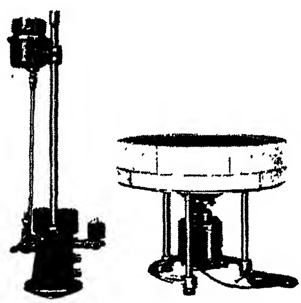
PETRO-NOKOL DOMESTIC OIL BURNERS

Models "W-1A," "W-1B," "W-1C," Horiztonal, Rotary Cup Type Burners



Model "W-1A"
Horizontal, Rotary Cup Type Burner

Model "W-1" is a development and a refinement of the giant burners now heating some of the world's largest buildings. It embraces every operating principle—every important detail of design—the same precision in manufacture—the same economy and reliability. It is simply smaller—a giant in "junior" size—and of course simplified. In Model "W-1" a long, conical, whirling rotary cup reduces heavy fuel oil to a thin film before it reaches the edge of the cup. Here the high speed of the cup's edge breaks the oil up into a mist. Air from the fan in the burner passes down through the nozzle of the burner—and out over the edge of the cup where it is deflected by angular vanes in a direction opposite to the rotation of the cup. Air and oil meet off the end of the rotary cup, moving in opposite directions, and form a highly turbulent combustible mixture which ignites instantly and burns in suspension into an incandescent hot gas. The exact amount of air for the amount of oil required is governed by an adjustment built in the burner.



Model "K": Rotary Burner

Model "K" Rotary Type Burner

The Model "K" is a fully automatic, forced draft, centrifugal atomizing burner for the small round boiler or furnace. Its design provides for complete control of the air for combustion over the oil consumption range. This assures maximum operating efficiency. The air and oil are delivered through a rotary head assembly revolving at a speed of 1750 r.p.m. Air and oil are thoroughly mixed in their passage across the deck installed to cover the grate area. Ignition, either gas or electric takes place slightly above the base of the sections of a refractory ring provided to aid combustion. Model "K" produces the brilliant luminous flame, characteristic of all Petro & Nokol Burners. It provides the radiant heat which most boilers are designed to take advantage of. This flame is concentrated at the grate level . . . . where the coolest water absorbs the maximum heat. Model "K" burners are built for operation with No. 2 oil or lighter, and have a capacity up to 2.1 gal. per hour.

Control and Safety Devices—All domestic and the smaller industrial burners are fully automatic in operation. All automatic burners are equipped with complete automatic control devices protecting both the burner and the boiler or furnace from abnormal operating conditions. Special controls and protective devices can be supplied to meet unusual operation conditions.

Undivided Responsibility—Throughout the Atlantic seaboard territory and in many of the more important mid-western centers, this company offers an undivided responsibility embracing the installation, servicing, provision of proper fuel oil, which insures satisfaction with Petro & Nokol equipment.

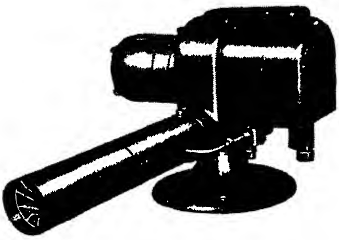
A factory field engineering organization is available at all times for survey, and conference in all things related to the use of oil as a fuel.

Performance and Acceptance—The parentage of the Petroleum Heat and Power Company extends back to 1903—to the development of the first modern industrial oil burner. In the intervening years more than 100,000 installations have been made from coast to coast. Today, Petro & Nokol oil burners are found in practically every type of building from the 52-story Metropolitan Life Insurance Building and the Ritz-Carlton Hotel in New York, to modern residences in suburban and rural districts.

A Few Prominent Installations

Equitable Building	New York City
Riverside Memorial Church	New York City
New York Hospital—Cornell Medical	New York City
Mark Hopkins Hotel	San Francisco
Standard Oil Company, 26 Broadway	New York City
American Central Life Insurance Company	Indianapolis
Pacific Telephone & Telegraph Building	Los Angeles
Sinclair Building	San Antonio
University of Toledo	Toledo, Ohio
Detroit Public Library	Detroit, Mich.

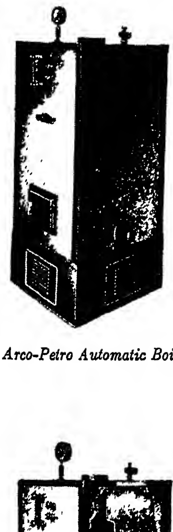
All Petro & Nokol burners are approved and listed by the Underwriters Laboratories, as well as by all other Government, State and Municipal authorities.



Model "P": Pressure Atomizing Burner

Model "P" Pressure Type Burners

A fully automatic, forced draft, pressure atomizing burners. Design provides complete control of the air required for efficient combustion over the oil consumption range assuring maximum operation efficiencies. The air is delivered to the combustion chamber through vanes arranged tangentially to the flow of air in the tube which impart to the air a centrifugal motion. This swirling stream of air mixes thoroughly with the oil atomized from the nozzle to form a lightly combustible compound. The quantity of air is regulated by means of the fan intake shutter. The Model "P" is a very simple mechanical device. The only moving part in the burner is a small motor, fan and oil pump, which combine as a single rotating unit. It runs on two large oversized bearings. Both static and dynamic balance of this rotor are easily secured. This insures quiet operation. It also eliminates the usual gear or belt drive. The flow of oil through the burner is controlled by a unique regulator that governs both minimum and maximum pressures. No oil is admitted to the combustion chamber unless there is adequate pressure for proper atomization of the fuel. An unusually effective strainer is built into the burner to remove any foreign substances from the oil that could possibly obstruct the flow of oil through the burner. When the Model "P" starts, a high tension electric spark instantly provides ignition for the atomized fuel. It continues until the burner stops. A momentary stoppage of oil has no effect because this continuous ignition is unfailing—always there ahead of any oil.



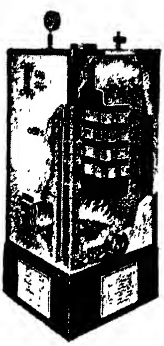
Arco-Petro Automatic Boiler

ARCO-PETRO AUTOMATIC BOILERS

Oil or Gas Burning Types

Arco-Petro Automatic Boiler—For Oil or Gas

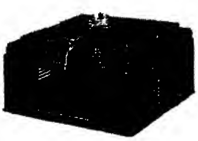
In addition to the oil burners listed in the accompanying table, Petro offers a series of domestic steam and hot water boilers containing built-in Petro oil burners or gas burners, in 6 sizes for 285 to 800 square feet of steam radiation, with corresponding capacities for hot water radiation. Conversion from oil burner to gas burner or vice versa is easily made by mere substitution of the desired burner assembly. Fuel consumption is exceptionally low because of the perfect co-ordination made possible by building burner and boiler for each other. Units are encased in heavily insulated, attractively finished cabinets of superior appearance. Yet the cost of this fine equipment completely installed is commonly no greater than the cost of other boilers when equipped with an oil burner.



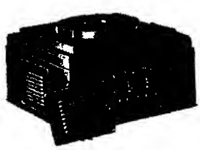
Arco-Petro Automatic Boiler

The Facts At a Glance

Commonly saves ⅓ to ½ on fuel.
Completely automatic—with oil or gas.
Petro Radiant Rotary Burner—or latest type Gas Burner.
Burner concealed inside boiler.
Quiet operation
Boiler especially designed for highest operating efficiency with burner.
A complete unit, ready to be connected to the piping.
For steam, hot water, or vapor.
Provides automatic hot water service at lowest cost—summer as well as winter.
Beautiful color combinations.
Costs no more than an ordinary boiler and oil burner.



Base and Burner for Oil



Base and Burner for Gas

Data and Dimensions

Boiler Model	T-11	T-12	T-13	T-14	T-2
Max. Cap. Total Tax Sq. Ft. Steam Rad.	285	390	495	600	800
Max. Cap. Total Tax Sq. Ft. H. W. Rad.	456	624	794	960	1280
Water Cap. to Water Line Gal.	15	17	19	21	16
Outlets—Number and Size	2-3″	2-3″	2-3″	2-3″	2-3″
Returns—Number and Size	2-3″	2-3″	2-3″	2-3″	2-3″
Shipping Weight—Pounds	1385	1435	1510	1675	1478

PETRO COMMERCIAL AND INDUSTRIAL OIL BURNING SYSTEMS

Model "W" Direct Driven Rotary Cup Type Burner

The Model "W" is a complete oil burning unit in itself, combining motor, fan, oil pump and atomizer, completely synchronized and controlled. This type of equipment is designed for medium-sized commercial and industrial plants and large, single boiler installations.

Model "H" Air Turbine Driven Cup Type

This equipment is adapted to medium and large boiler plants with multiple boiler installations, also for industrial applications. Burns all grades of fuel oil. Low pressure air and oil are supplied from the combination fan and pump set to a series of burners and air registers mounted at the boiler fronts. Operating burners from a central source of motive power eliminates the multiple installation of electrical equipment. Fan and pump sets can be driven by electric motor, steam, turbine, or gasoline engine power. Additional firing units can be added at any time without increase of motorized equipment. Low pressure air is delivered through an air line buried in the floor. Secondary air is drawn through the controlling air vane registers by natural draft.

Model "M" Mechanical Type

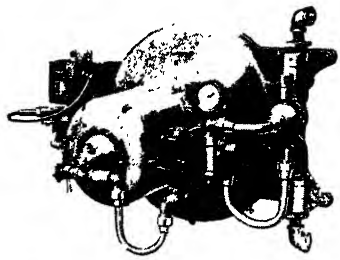
Model "M" is now adaptable for either natural or forced draft in one standard design. Oil is delivered to the burners at high pressure and high temperature (approximately 250 pounds and 250° F.) and atomized through a small orifice at the improved tip. The oil is supplied by steam pumps.

Petro Oil Pumps for All Services

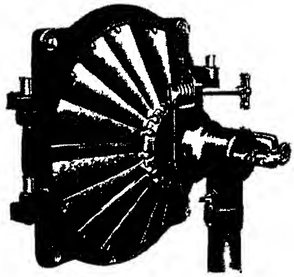
The Petro Belt Driven Pump motor operates at a full load speed of 1,750 r.p.m. and is of ample capacity to meet load demands. Motors are available for use with alternating or direct current. Pump speed is 220 r.p.m. Bunker "C" oil may be handled as readily as furnace oil.

The Petro Domestic Wall Pump is designed for installations where fuel oil is fed to the burner by static pressure created by a column of oil. The reservoir maintains a steady head of oil. Available in two sizes, rated at 10 and 20 gallons of No. 3 oil per hour.

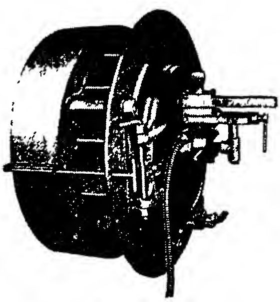
The Petro Worm Drive Pump is a heavy duty pump designed for installations where regulations impose the most exacting requirements.



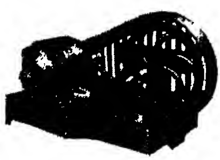
Model "W": Horizontal, Direct Motor Driven Rotary Cup Type Burner



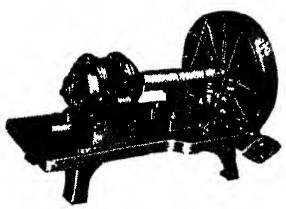
Model "H": Air Turbine Driven Rotary Cup Burner



Model "M" Mechanical Type



Petro Belt Driven Pump



Model "H" Combination Fan and Pump Set

Iron Fireman Manufacturing Company

Automatic Coal Burners

Portland, Oregon

Factories: Portland, Ore.; Cleveland, Ohio; Toronto, Canada

Retail Branches or Subsidiaries

Chicago, Ill. Milwaukee, Wis. St. Louis, Mo. New York, N. Y.

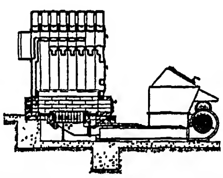
Dealers in Principal Cities and Towns in the United States and Canada

Representation in numerous foreign countries.

IRON FIREMAN Automatic Coal Burners

"Forced Underfiring" Principle—Iron Fireman "Forced Underfiring" is based on the scientific principle of feeding fuel to the fire from below, under forced draft. From the conveyor screw, coal enters the firebox under the fire and is gradually forced upward into the flame. As the coal approaches the fire, it is gradually heated. The volatile gases are distilled off in the presence of an excess of oxygen and are thoroughly ignited while passing through the incandescent fuel bed. This insures complete combustion. The ash is fused into clinkers which are easily removed.

Advantages—Iron Fireman saves money and increases heating plant efficiency in four major ways: (1) Cuts fuel costs; (2) reduces labor costs; (3) provides steady, even heat or power; (4) eliminates the smoke nuisance.

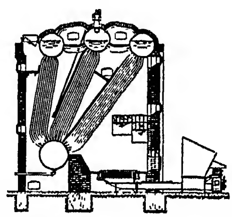


Typical Iron Fireman Installation under Cast Iron Boiler

Installation and Sizes—Iron Fireman is made in a range of sizes for commercial heating and power plants up to 250 h.p. boilers, and also for homes. It can be installed quickly in practically any solid fuel boiler or furnace, old or new. If necessary, the installation can be made with practically no interruption of the service from the boiler.

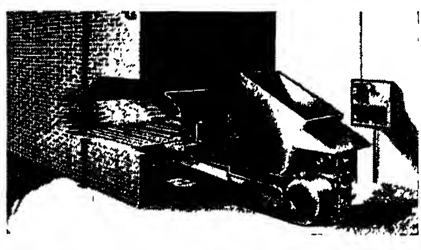
Machines are shipped complete from the factory. All parts are standard and interchangeable.

Features of Design and Construction—Construction and operation of the Iron Fireman are characterized by simplicity throughout. Outstanding features of design and construction are: (1) Pressed steel construction. (2) Special patented transmission—three speeds and neutral. (3) Continuous feed transmission. Gears run in bath of oil. (4) Electric motor—standard make. (5) V-belt drive. (6) Safety shear pin protects mechanism from damage. (7) Quiet ball bearing fan supplying forced draft to fire. (8) Automatic fire banking damper—conserves fuel and holds fire in proper condition when stoker is idle. (9) Positive pneumatic fume eliminator—an auxiliary air supply that insures positive movement of all gases through the fire. (10) Sectional retort especially designed to allow for heat expansion. (11) Dead plates of heavy iron and ribbed. (12) Sectional, self-cleaning tuyere blocks. (13) Conveyor screw cast of special Iron Fireman alloy steel from one-piece pattern. (14) Automatic electric controls, designed for and used exclusively on Iron Fireman.



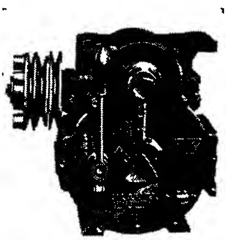
Typical Iron Fireman Installation under Four Drum Water Tube Boiler

Iron Fireman in Operation. Installed under Horizontal Return Tubular Boiler. Low Bridge Wall.



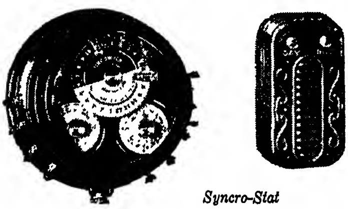
Cutaway View, showing details of Typical Iron Fireman Installation

Transmission and Continuous Feed Principle—The Iron Fireman speed reduction unit and its patented speed-change gears are an exclusive Iron Fireman development. This transmission drives the conveyor screw at a constant speed which in turn feeds the coal to the fire in a slow, steady stream at the required rate for proper burning. As a result a steady, non-agitated fire is obtained. The Iron Fireman transmission has three speeds and three neutrals. The gears can be shifted while the stoker is in operation; in fact more easily than when the stoker is idle, and it is impossible to strip gears while shifting. All the gears operate in a bath of lubricating oil and the unit is most quiet. In design, materials and precision of construction, the Iron Fireman transmission is like that of a fine automobile.



Iron Fireman Transmission

Automatic Controls—Iron Fireman starts and stops at the command of sensitive, accurate automatic controls. Directing controls govern stoker operation according to demands of time, temperature, or pressure. An example of the superiority of Iron Fireman directing controls is the "Syncro-Stat" which provides automatic



Syncro-Stat

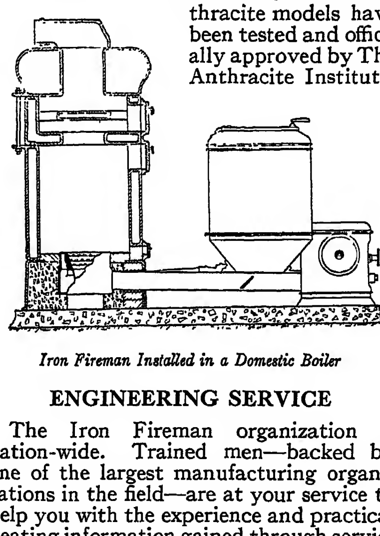
control of day and night temperature. It is powered by a Telechron motor, thus eliminating winding or setting, and making it an accurate all-electric temperature regulator. Other directing controls include pressure regulators, hot water and furnace regulators and the "Timetactor" a device which runs the Iron Fireman during predetermined intervals in order to keep the fire alive during mild weather.

The most important unit of the operating control system is the motor-driven relay switch. This device starts and stops the stoker motor at the command of the Syncro-Stat or other directing controls. In the case of the larger stokers a magnetic operating switch works in conjunction with the relay switch.

Important Factor in Plant Economy—Iron Fireman users report savings of from 15 per cent to 50 per cent on fuel costs alone. In a recent investigation of 342 users, fuel savings alone averaged an earning of 39.44 per cent a year on the total cost of their installations.

Iron Fireman for Homes—The Iron Fireman residential model employs "Forced Underfiring" principle the same as larger machines, with simplified operation. The enclosed hopper accommodates sufficient small size coal for an ordinary day's consumption.

Can be recommended for any steam, hot water, vacuum or warm air furnace. Quickly installed. Models for both bituminous and anthracite coal. All anthracite models have been tested and officially approved by The Anthracite Institute.



Iron Fireman Installed in a Domestic Boiler

ENGINEERING SERVICE

The Iron Fireman organization is nation-wide. Trained men—backed by one of the largest manufacturing organizations in the field—are at your service to help you with the experience and practical heating information gained through servicing thousands of boiler rooms and heating plants in all parts of the country.

Any Iron Fireman engineer will gladly call and submit any additional information requested.

CATALOG AND INFORMATION

Catalog No. 33 gives full information about the Iron Fireman. Descriptive folders give special data about installation in particular types of industries and in homes.

Secure them by addressing the factory or any Iron Fireman representative.

Detroit Stoker Company

Sales Offices and Engineering Department — General Motors Bldg., Detroit, Mich.

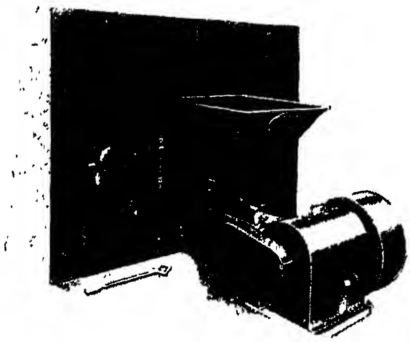
Main Office and Works at Monroe, Mich.

Since 1898

District Offices in Principal Cities

Built in Canada at London, Ont.

A Detroit Stoker for Every Service, and all sizes of boilers, from small heating boilers, to large water tube boilers. Each installation, regardless of size is carefully studied from an engineering standpoint to insure best results.



Detroit LoStoker, showing Motor Driven Blower and Stoker, One Compact Unit at the Front

Detroit LoStoker Advantages include:

Agitator in coal hopper for positive coal feed. Cannot stick or jam with wet coal.

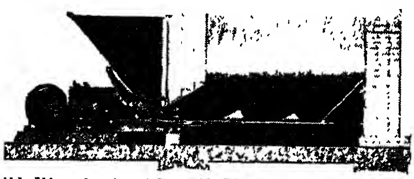
Adjustable Plunger Feed for the control of the quantity of coal and its distribution within the furnace.

Heavy Mechanical Drive of simple design and with Timken Bearings. Little power is required for operation.

Side Cleaning with dumping grates. Ash doors are provided. No hand cleaning.

Automatically Controlled. Detroit LoStokers are motor or turbine driven, controlled from steam pressure, water temperature or room thermostat.

Write for Bulletin 369.

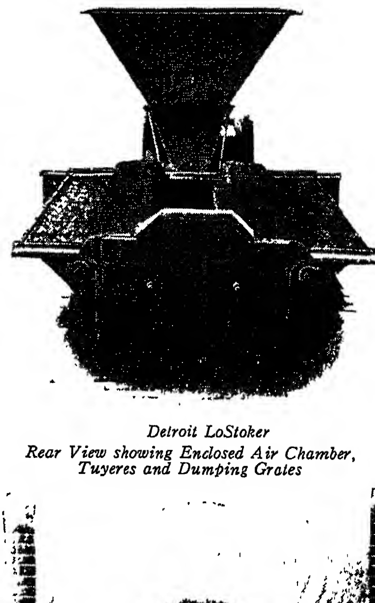


Side View showing Adjustable Plunger Feed. No Moving Parts in or Near the Fire

Detroit LoStoker

Built in various widths and lengths to fit furnaces of all types of boilers. Compact, easily installed, responsive and automatic. Savings, due to increased efficiency, combined with the ability to successfully burn less expensive grades of coal, make Detroit Stokers pay a handsome return on the investment.

Write for Bulletin 369.



Detroit LoStoker
Rear View showing Enclosed Air Chamber, Tuyeres and Dumping Grates

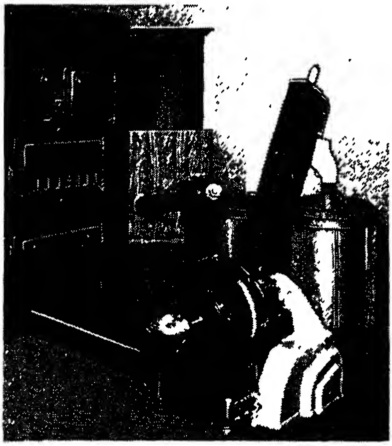
Front Sectional View. Large Active Fuel Bed, with Provision for Admitting Air under the Dumping Grates at each side to burn out the Combustible Prior to Dumping Ashes

290
Hudson Street

MOTOR STOKOR
CORPORATION

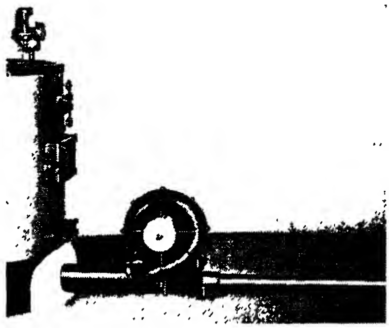
New York
N. Y.

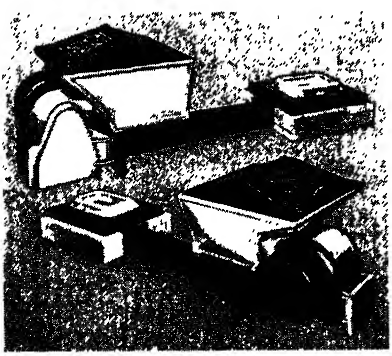
A FULLY AUTOMATIC COAL BURNER



Series 2AF Motorstokor, the ash removing — anthracite — universal type feeder model, conveys "buckwheat" or "rice" coal direct from the bin into an underfeed retort. Ashes are removed into two or seven full size ash barrels. Minneapolis-Honeywell series 10 controls are standard equipment on all Motorstokors.

Series 5AF Motorstokor, the straight line feeder model offers automatic coal feed from the bin with ash removal by gravity into a sunken pit.





Series 3BH Motorstokor is designed for bituminous coal. The remarkable retort design, originated in Motorstokor's research laboratory, enable 3BH models to burn even high volatile, caking and coking "slack" without creating smoke and without manual agitation.

Other bituminous models provide automatic coal feed from the bin and automatic ash removal.

Please ask for catalog and specifications. 60 models and sizes from ten lbs. per hour to 200 lbs. per hour feed rates. (300 to 7,000 sq. ft. E.D.R. on single units, up to 140 H.P. on multiple installations.

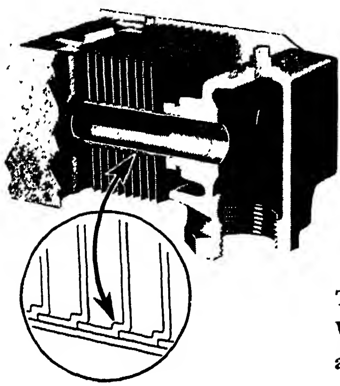
Chase Brass & Copper Co.

INCORPORATED

Heating Products Division

Waterbury **Connecticut**

For data on Chase Copper Tube and Sweat Fittings for Heating Lines see pages 696-697



Chase Copper Radiators

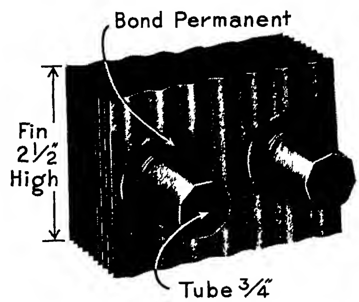
The one type can be used for Hot Water, One and Two Pipe Steam, and Vapor Vacuum Heating Systems

CONSTRUCTION—Copper, the best commercially practicable material from a heat transfer standpoint, is used for both the tube and fins. The tubes are firmly secured to the headers by compression nuts, a universally accepted method of making a permanent bond not affected by strain or stress. The fins when they contact the tubes, have a flange or lip, which acts as a definite spacer for the fins. Part of the lips are lapped over and under each other to prevent slipping and insure a tighter bond on the tube. Fins are corrugated to provide greater area for a given depth of unit and to increase strength and rigidity.

TUBE SIZE—On the Chase copper radiator the supply tubes are ¾ in. diameter. This is an important advantage as it assures a more satisfactory operation. For example, when used with hot water it permits freer circulation. On one pipe steam installations there is enough room for condensation return.

ADVANTAGES—Here are four advantages of the Chase Radiator. **1.** The use of copper and brass materials with high heat conductivity and long life. **2.** Adequate and permanent bond between fin and tube and between tube and header. **3.** Sufficient steam on waterway to meet all conditions of service. **4.** High heat output with respect to space required and to weight.

Complete catalog filed in Sweet's Architectural Catalog



The copper to copper bond is accomplished by a patented process of tube expansion. No solder is used

Over Fifty Years In Business

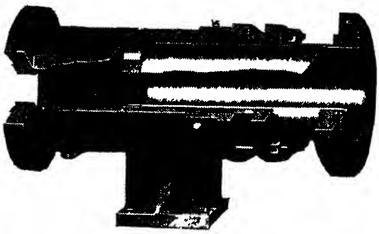
AMERICAN DISTRICT STEAM COMPANY

GENERAL OFFICES AND WORKS
NORTH TONAWANDA, N.Y.

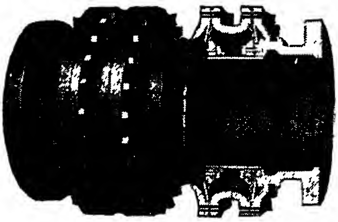
STEAM DISTRIBUTION EQUIPMENT

Branches and Agents in Principal Cities

ADSCO EXPANSION JOINTS



Internally Guided Slip Type



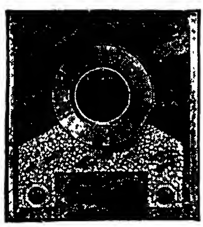
Packless Variator

ADSCO Expansion Joints, whether packless or slip tube type, are available in a complete line of types and sizes; for all pressure to 400 pounds and temperatures to 750°F.

Slip tube types range from the simple semi-guided joint to the completely guided type with all bearing surfaces of bronze and no possible metal to metal contact between any part of the joint and the sliding surface of the slip, a feature obtainable only in ADSCO Joints, and one of the reasons leading architects and engineers specify them.

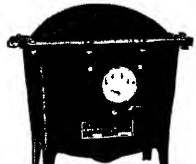
ADSCO Packless Expansion Joints (Variators) utilize a flexible stainless steel diaphragm to control expansion and contraction. Movement is controlled both ways. There are no strains on the flexible members because all stresses are transmitted to the joint body by backing plates—a most unique and successful method as proved by the hundreds installed more than 25 years ago, with no maintenance since installation. Engineers are constantly giving them their stamp of approval for use where accessibility is practically impossible when the job is completed.

ADSCO UNDERGROUND CASING CONDUIT



ADSCO Red Diamond Brand Casing—an ideal underground steam or hot water line conduit since it is both conduit and insulation—is real economy for a college or institution distribution system. Doing service after 38 years is conclusive evidence. More than 90 per cent efficient. Furnished in split form for two or more pipes, if desired. Lays quickly because of 5 to 8 ft. lengths. Trench may be backfilled immediately. Does not expand or contract longitudinally, assuring permanent water-proof construction.

ADSCO ROTARY CONDENSATION METER



For measuring the steam condensation of a heating system or heating equipment. Commercial distributors of steam use them as a basis of charge for steam sold—institutional groups for steam cost distribution—and industrial firms for steam consumed in process work. Accurate within 1 per cent of absolute. Dependable. Compact. Reads directly in pounds of condensed steam. Furnished with cast-iron or aluminum body and cover in seven sizes: 250, 500, 750, 1500, 3000, 6000 and 12,000 pounds capacity per hour.

E. B. Badger & Sons Co.

63-75 Pitts Street, **Boston, Mass.**

Engineers and Manufacturers

NEW YORK OFFICE, 271 Madison Avenue — OFFICES IN PRINCIPAL CITIES

PRODUCTS: Corrugated Expansion Joints, Pipe Bends, Chemical Apparatus, Copper and Sheet Metal Work, Copper Boilers and Hot Water Tanks.

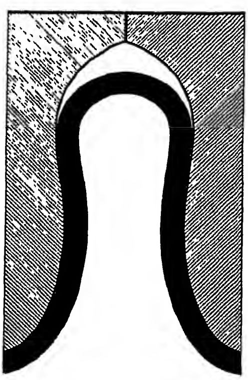
BADGER SELF-EQUALIZING EXPANSION JOINTS

These have been on the market for more than forty years and as sold today represent the results of a series of distinct improvements. Badger engineers are responsible for such developments in the corrugated type of joint as special analysis copper of which the seamless tubes have been made, the equalizing rings, the monel metal sleeve to protect against superheated steam, the welding end joint. Now, Badger Joints carry another big improvement in a new design corrugation which functions on the principle of Directed Flexing.

Before passing to this subject, a word about the importance of using the corrugated type of joint. Not only is it compact and easily installed, but it requires no maintenance. It uses no packing.

Directed Flexing

The shape of the new Badger Corrugation is entirely curved. There are no straight sides. In action the corrugation undulates, using the equalizing ring as a guide. This is the so-called "Directed Flexing." Instead of being left to chance, the flexing of the metal is guided so as to undulate along an all-curved surface. This is well brought out in the two sections shown. The shaded portions are two halves of adjacent equalizing rings; the heavy line is the corrugation. Note how it is wrapped around the curve of the ring.

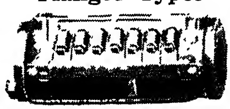


Longer joint life is the direct result of this improvement. A new bulletin, describing this new corrugation, will be sent on request.

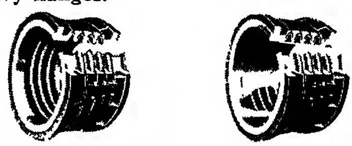
Different Types Available

Both flanged end and welding end joints are available from 3 in. (welding) and 4 in. (flanged) up. Single or double joints with or without service outlets. All joints can be equipped with telescoping monel metal sleeves to protect against superheated steam.

Flanged Types



Fitted with alignment bars on the 4 and 5 in. sizes. Choice of standard or extra heavy flanges.



On 6 in. pipe and larger, no alignment bar is used. Standard 125 lb. or extra heavy 250 lb. flanges as required.

Welding Types

For use with both saturated and superheated steam. Single type from 3 in. up, to take care of 1 in. expansion or more. Pipe nipples welded directly into pipe line.

Double units for 2 in. expansion or more. Equipped with service outlets as shown, if desired. Complete units furnished, mounted on base plate, with anchor and guides.

Single and Multiple Corrugated Expansion Joints for Low Pressure

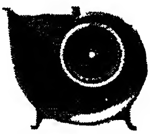
For use between turbine or engine exhausts and condenser or on low pressure lines. Excellent for absorbing shock and vibration. Flanges in round, oval or rectangular shapes. Guaranteed up to 30 lb. pressure.

Bayley Blower Company

1817 S. Sixty-Sixth Street **Branches in Principal Cities** **Milwaukee, Wis.**

Builders of Heating, Ventilating, Cooling, Purifying, Humidifying and Air Washing Equipment; Exhaust and Drying Apparatus, Mechanical Draft and Blast, Fans and Blowers of all Types

Bayley Plexiform Fan:



Is a multi-blade fan for supplying air for heating and ventilating systems, manufacturing processes, drying systems, forced and induced draft systems. It is suitable for handling high or low temperature gases at medium or low pressure. Will deliver maximum quantities requiring minimum space with great economy.

This is a distinct Bayley product, high class material and workmanship, properly designed to avoid excessive vibration and overstressing of parts. Inlets and outlets are properly sized for maximum delivery and maximum efficiency. Fans are furnished in single or double width of any required arrangement and with sleeve or anti-friction bearings.

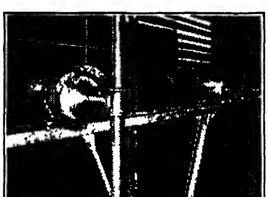
Aeroplex Fan:

Is of high speed design with self limiting power characteristics. Application parallel to the Plexiform Fan. Highly efficient and quiet in operation.

Bayley Exhausters and Pressure Blowers:

Type "B" exhaust fan is for heavy duty, handling refuse from industrial and textile plants. Type "SE" is used in handling smoke, fumes and dust-laden gases. Type "H" for high-pressure work. These units are highly efficient and of high class design and workmanship.

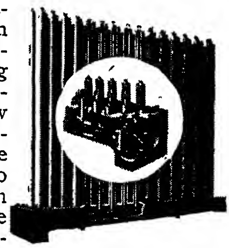
Bayley Turbo Air Washers, Humidifiers and De-Humidifiers:



The Bayley Turbo Air Washer Showing Turbo Atomizer and Eliminator

The Turbo Atomizer used in the Bayley Washer produces a steady, fine spray. Water at low pressure is delivered to the center of a rapidly revolving cone-shaped rotor provided with atomizing pins set in its periphery. This atomizer requires very little attention, and will operate successfully under low water pressure. The orifices are large and this atomizer, unlike high pressure nozzles, cannot clog.

Bayley Chinook Heating Sections:



The Chinook section is used with blast heating, ventilating and drying systems, and is suitable for high or low pressure steam circulation. The base is divided into two chambers. Steam enters (see cut) the lower chamber, rising through ⅜-in. pipes located within the 1¼-in. pipes leading from the upper chamber. Condensation takes place in the larger pipes, the water falling into the upper chamber and draining away through the return outlet. The Chinook can be repaired in the middle of the bank without breaking steam connections or taking down a section.

Shipped assembled in smaller sizes, and knocked down in the larger units. May be installed in horizontal or vertical position.

Bayley Chinookfin Heating Sections:

Are the same design as the Chinook Heaters, using heavy gauge copper fin tubes. As compared with Chinook it is much lighter and occupies less space.

Bayley Plexfin Unit Heaters:

This unit incorporates Chinookfin radiation and Plexiform or Aeroplex fans. The fan assembly including top plate and motor is removable as a unit for maintenance and inspection. The heating element is a removable unit. Casing all welded extra heavy gauge. This is an exceptionally high grade unit at a moderate price.

Buffalo Forge Company

484 Broadway, **Buffalo, N. Y.**

Branch Offices

ALBANY, N. Y. ....414 Standard Bldg., H. S. Johnson
ATLANTA, GA. ....404 Title Bldg., E. T. Gorbandt
BALTIMORE, MD. ....303 Morris Bldg., R. W. Hogan
BOSTON, MASS. ....P. O. Box 71, E. D. Johnson
CHARLOTTE, N C. ....P. O. Box 376, J. W. Fraser
CHICAGO, ILL. ....15 N. Jefferson St., L. D. Emmert
CINCINNATI, OHIO ....622 Broadway, F. W. Twombly
CLEVELAND, OHIO ....418 Rockefeller Bldg., T. A. Weager
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DENVER, COLO.,
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GREENVILLE, S. C. ....P. O. Box 563, G. R. Morgan
HUNTINGTON, W. VA. ....1612½ Third Ave., J. K. Giffen
INDIANAPOLIS, IND. ....725-727 Circle Tower, I. C. DeHaven
KANSAS CITY, MO. ....217 Dwight Bldg., L. R. Chase
KIRCHENER, ONT., CANADA,
Canadian Blower & Forge Co., Ltd., A. S. Capwell
KNOXVILLE, TENN. ....P. O. Box 1035, C. F. Sexton
LOS ANGELES, CALIF.,
610 Pershing Square Bldg., P. R. Adrianse
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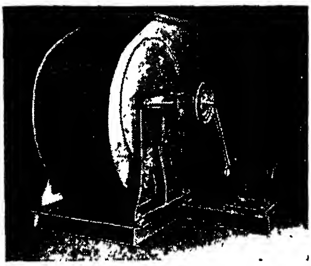
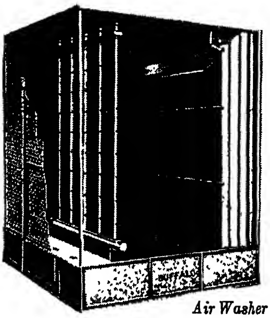
NEW YORK, N. Y. ....39 Cortlandt St., W. S. Koithan
NEW ORLEANS ....2841 Pine St., A. C. Hays
PHILADELPHIA, PA. ....220 South 16th St., L. C. Davidson
PITTSBURGH, PA. ....912 Fulton Bldg., H. L. Moore
SAN FRANCISCO, CALIF.,
Herberts Moore Machinery Co., 550 Fifth St., J. G. Scott
ST. LOUIS, MO. ....1596 Arcade Bldg., J. W. Cooper
SALT LAKE CITY, UTAH,
Salt Lake City Hdw. Co., W. H. Trasls, Jr.
SEATTLE, WASH. ....2719 First Ave. S., A. T. Forsyth
TOLEDO, OHIO ....1817 North 13th St., C. M. Eyster
WASHINGTON, D. C.,
Commercial National Bank Bldg., G. S. Frankel
WILKES-BARRE, PA.,
Power Eng. Corp., Coal Exchange Bldg., C. Ide

PRODUCTS—Heating and Ventilating Equipment, including: Unit Heaters, Multiblade Fans, Pipe Coil Heaters, Buffalo Air Washers, Buffalo Unit Air Washers, Buffalo Unit Coolers, Drying Equipment, Mechanical Draft Fans, Air Preheaters, Exhaust Fans, Blowers, Dust Collectors, Disc Fans, Spray Nozzles.



Buffalo "Limit Load" Conoidal Fans with Silent Floating Base—The quiet, high-efficiency "Limit-Load" Conoidal fan is now available on the silent floating Buffalo Base which is rubber-insulated so that no motor or fan vibrations are transmitted to structural frame work of the building. This makes an almost noiseless installation, particularly suitable for auditoriums, theatres, churches, hotels, etc.

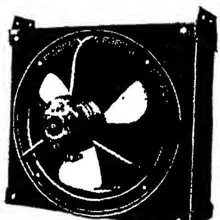
Buffalo Air Washers



Air Washer

Developed from the original design, Buffalo Air Washers have demountable, one-piece eliminators, non-clogging spray nozzles, fully flooded scrubbing surfaces, and wide suction screen.

Gas and Steam Unit Heaters



Breezo-Fin

Both types of steam unit have extended surface copper heating coils.

Buffalo Gas Unit Heaters are made in eight sizes with capacities from 75,000 B.t.u. to 500,000 B.t.u. input per hour. They are suitable for burning any kind of gas, natural or manufactured, and provide unusual heating economy.

Breezo Ventilating Fans

In sizes from 8 in. to 36 in. diameter, Breezo fans provide inexpensive efficient ventilation on any job where they may be used to exhaust into the open. Popular 8 in. home ventilating model is supplied in metal case for installation in kitchen wall.

Unit Coolers

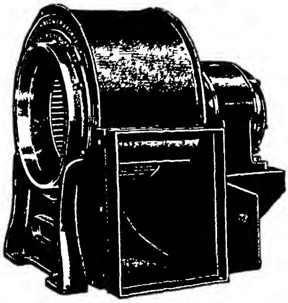
Buffalo Unit Coolers are made in floor and suspended types for use with ammonia, brine, methyl chloride or freon refrigerants. They are suitable for installation in chilling rooms, packing plants, fruit storage rooms, etc. Complete information contained in Bulletin 2904.

Champion Blower & Forge Co.

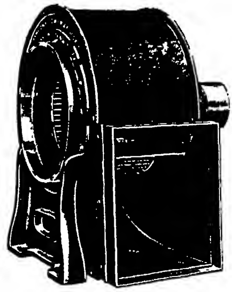
Manufacturers and Engineers

Plant and General Offices: **Lancaster, Pa.**

Manufacturers of Blowers, Ventilating Fans, and Exhaust Fans for Air and Material; Unit Heaters and Blast Gates.



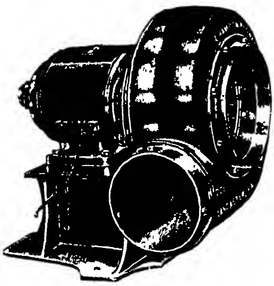
Type "SE" Electric Driven Fans



Type "S" Manivane Belt Driven Fans

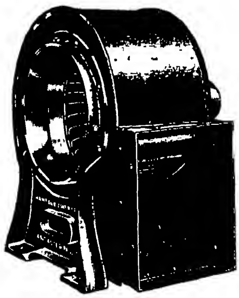
Sizes—6″ to 36″ Wheels

For Heating, Ventilating, Cooling, Drying or Forced Draft



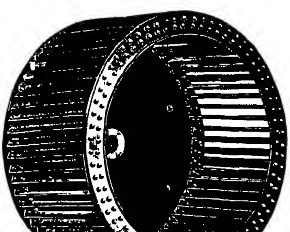
Type "CE" Manivane Fans

For Duct Work and Quiet Operation.



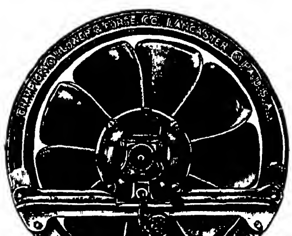
Type "BC" Backward Curve Fan

Sizes—12″ to 72″ Wheels



Type "C" Manivane Blast Wheel

Built Strong and Substantial. Free from Vibration and Noise. Very desirable for Oil Burners and General Heating and Ventilating Purposes, especially where noise is an objection.



Super Fans for Free Air

To Ventilate and Remove Odors or Foul Air from Rooms, Shops, Mills, Factories, Foundries and Steam in Laundries and Steam Rooms.

Sizes—12″ to 36″.

L. J. Wing Mfg. Co.

Branch Offices in Principal Cities

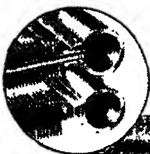
59 Seventh Avenue, **New York**

PHONE: CHELSEA-3 0027-0030

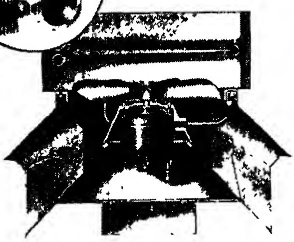
Factory: NEWARK, N. J.

Wing Featherweight Unit Heaters and Process Heating Units; Utility Unit Heaters; Wing Scruplex Safety Ventilating Fans, Fog Eliminators and Exhausters; Wing Forced Draft Blowers, Turbine or Motor-Driven; Steam Turbines; Man-Coolers.

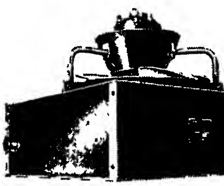
LIGHT weight, and vertical downward discharge of the heated air through high velocity multiple discharge outlets are original and unique features of Wing Featherweight Unit Heaters. Thus, downward circulation of large volumes of warmed air to the floor and uniform distribution of it over the entire area is assured, with resultant economy in plant heating.



Detail of Wing Featherfin Heating Element showing Compression Union Tube Connection



TYPE LC UNIT
Cross-Section showing location of motor and fan. Used in buildings having low roofs or ceilings



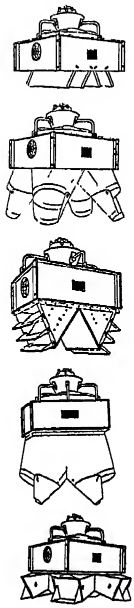
TYPE HC UNIT
Complete coverage accomplished with any of discharge outlets illustrated at right

Wing Featherweight Unit Heaters produce a pleasing sense of warmth at the floor level because the warmed air from each of the several heater discharges actually reaches the floor. On the other hand, attempting to accomplish this result by the method of withdrawing cold air from the floor would create cold drafts and circulate floor dust through the atmosphere.

Wing Featherfin Heating Element—(Tested to 1000 lbs. Pressure). Extremely light in weight. By simple variations of the heating surface any desired final air temperature may be obtained with any given steam pressure.

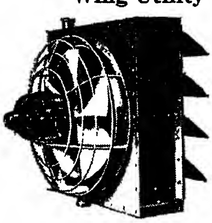
Fin and tube extended surface type. Hairpin or return bend design. Headers of steel, tubular design. Tubes secured to headers by steam-tight compression union. Easily removed and replaced in case of damage.

Standard Discharge Outlets for Type HC



BULLETIN H-5 CONTAINS COMPLETE DATA

Wing Utility Unit Heater

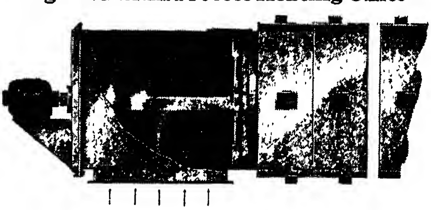


A general purpose heater. Delivers heated air in one general direction. With vane diffusers and safety guard for fan. Capacity data same as Type HC. *Bulletin U-3.*

Wing-Scruplex Fog Eliminators

Supply tempered fresh air to completely de-fog any building or room where steam, fog or fumes are liberated in manufacturing processes—*e.g.*, Dye Houses, Creameries, Galvanizing Plants, etc.

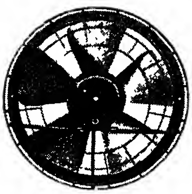
Wing Featherfin Process Heating Units

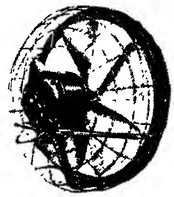


For manufacturing processes such as drying, ageing, etc., requiring the recirculation of the heated air. Motor or turbine located outside air current. *Bulletin P-2.*

Wing Featherfin Heater Sections for general blast heating. Bulletin HS-1.

Wing-Scruplex Safety Ventilating Fans



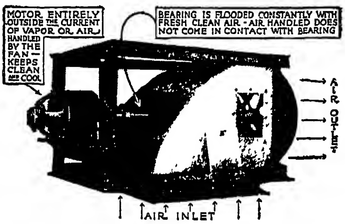


An integral guard of strong steel rings welded to the frame features every model of the Wing-Scruplex Safety Ventilating Fan. This guard removes the ever-present danger which exists whenever fans are in an exposed location, and does not reduce fan efficiency.

The high volumetric efficiency of Wing-Scruplex Safety Fans is due to their true screw propeller design which moves the air forward in straight lines without eddy. Motor is fully enclosed, easily accessible and generously proportioned. Can also be supplied with pulley or turbine drive.

Wing-Scruplex Safety Fans are built in the following sizes: 10, 13, 17, 22, 25, 30, 36, 42, 48, 54 and 60 in.—capacities from 950 c.f.m. to 33,000 c.f.m. Up to 25 in. dia. propellers are made of cast-aluminum—large sizes of pressed steel.

BULLETIN F-5 CONTAINS COMPLETE DATA



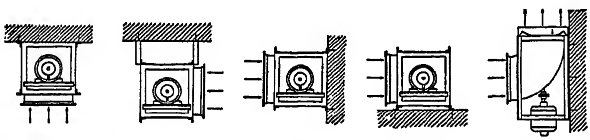
Wing-Scruplex Exhauster with bottom inlet

Wing-Scruplex Exhausters

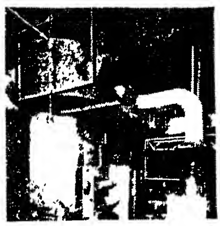
In the Wing-Scruplex Exhauster the motor is entirely outside the exhaust housing, therefore always easy of access, clean and cool. As an elbow in any duct system where resistance is moderate this exhauster provides efficient, low cost, convenient ventilation. For acid laden air, fan and exposed parts can be supplied in Monel metal.

Wing-Scruplex Exhausters are made for either horizontal or vertical operation—top, bottom or side intakes. The drawings below show the flexibility of application of these units.

BULLETIN E-8 CONTAINS COMPLETE DATA



Bolted directly to Ceiling — *Hung from Ceiling* — *Bolted directly to Side Wall* — *Bolted to Floor or Foundation* — *Vertical Bolted to Side Wall*



Wing Blowers for Forced Draft

Wing EM Blower

The installation of Wing Motor-driven Type EM Blowers on heating boilers of all sizes makes possible the use of Buckwheat coal and other inexpensive fuels with great savings in fuel costs, which usually pay for the blowers in the first year of operation.

These blowers are of the propeller type fan construction. They afford large air passages and low air velocities, resulting in quiet operation and even fires.

Wing Blowers controlled by speed regulation, conveniently operated from the front of the boiler. Totally enclosed motors keep out the dust and dirt of the boiler room, insuring many years of service without repair. Automatically controlled, they make the fireman's work easier.

Ask for Bulletin No. M-56 describing Wing EM Units. For High-Pressure Boilers ask for Bulletin No. T-97 describing Wing Turbine Blowers.

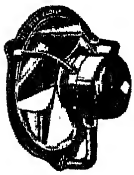
The Wing Blower in foreground supplied forced draft to hot water supply heater; while blower in background serves two return tubular heating boilers.

ILG Electric Ventilating Company

Propeller Fans, Blowers, Unit Heaters, Unit Coolers

2880 North Crawford Avenue, **Chicago, Ill.**

Sales Representatives in all Principal Cities



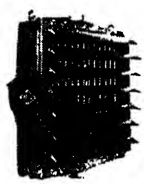
Ilg Self-Cooled Motor Propeller Fans

Furnished with direct-connected fully enclosed and self-cooled motor. Wheels rotational static balanced. Sizes 12 to 72 inches. Capacities 750 to 40,500 cubic feet per minute. Self-Cooled motor makes fan especially effective in handling gases and extreme heat. Used everywhere for general exhaust ventilating.

Ilg Universal Multiblade Blowers

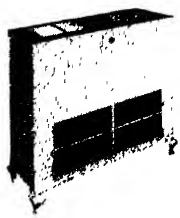
Ilg Type B Universal Blowers are designed so as to combine compactness, high efficiency, quietness and low power consumption. The motor is recessed in the side of the blower requiring no separate base and insuring quietness of operation, economy of installation. Ilg Blowers are furnished either direct connected to Ilg ballbearing motors or for belt drive. Sizes 25 to 90 single or double width.

Ilg Unit Heaters—Steam and Electric



Ilg built throughout with copper tube and fin coil and patented fully enclosed, self-cooled motor. For operation on steam or hot water. Tested with 500 pounds hydrostatic pressure. Available in 14 capacity sizes. Ilg electric unit heaters for all electric operation, available in 20 sizes for wall or ceiling suspension and for floor mounting.

Ilg-Kold Electric Cooling Systems



A new and simpler system of unit cooling for stores, offices, factories, restaurants, homes, etc. Floor cabinet or ceiling suspension units. Combines air cooling and air dehumidifying to obtain better hot weather air conditions. Installation can be made with a minimum of expense and time. Quiet and extremely effective.

OTHER ILG PRODUCTS
Kitchen Ventilators, Power Roof Ventilators, Humidifiers, Automatic Shutters, Ilg Columnar Heat Systems, Ilg-Kold Compressors.

ILG

For Offices, Stores, Factories, Restaurants, Theatres, Public Buildings, Homes, Etc.

VENTILATION

AEROFIN CORPORATION

850 Frelinghuysen Avenue
Newark, N.J.

Manufacturers of AEROFIN

THE STANDARDIZED LIGHT-WEIGHT FAN SYSTEM HEAT-SURFACE

11 West 42nd Street, NEW YORK

Land Title Building PHILADELPHIA	United Artists Building DETROIT	Burnham Building CHICAGO

Aerofin is the modern Standardized Light-Weight Fan System Heat-Surface originated by *Fan Engineers* to meet the present-day requirements of this highly specialized field, and to afford an adaptability which permits and fosters the new and advanced applications of tomorrow.

Flexitube AEROFIN

FLEXITUBE AEROFIN: (Fig. 1) supplants the former Low Pressure AEROFIN (the original light-weight, non-corrodible, encased Heat-Surface) and will, we believe, eventually supplant Universal AEROFIN, as Architects and Engineers become familiar with its unusually adaptable characteristics.

Flexitube AEROFIN is distinguished from all other developments by its off-set tubes,

Fig. 1

as shown. This ingenious idea imparts to the tube a flexibility which enables it to expand or contract, under temperature changes, without strain upon itself, the header joints, the header or the casing. Hence the name Flexitube AEROFIN.

The flexibility of the off-set tube permits a single-pass, header-to-header design which allows the use of rigid headers and so perfectly relieves or absorbs expansion and contraction strains that header and header-joint construction adequate for high pressure service may be employed, and the permanency of the construction assured thruout the practically unlimited life of the surface.

Single-pass tube construction is essential for low pressure service and is desirable for intermediate pressure service (up to 200 lbs.) since it insures uniform steam or cold water distribution and proper, quiet drainage of condensate, while permitting installation of the Unit *in any position*, The off-set tube of Flexitube AEROFIN is the first definite achievement of the ideal single-pass, header-to-header design. Tests equivalent to a life-time of service show that Flexitube AEROFIN will stand up indefinitely against the strains of expansion and contraction because the Unit, due to its design, does not *resist* these strains, but absorbs them in its flexible tubes without rack or injury.

The headers are one-piece brass castings, machined for tube joints and pipe tappings.

The joints between the tubes and the headers are of a patented type made by using a brazing material applied at high temperature and have proved stronger than the tube itself under all operating conditions for which AEROFIN is sold. We believe this joint to be the most practical and dependable yet developed.

Steam tappings are located on the center line of the casing with respect to its width, off-set from center line with respect to its depth. Return tappings are eccentric, at opposite end of casing.

This arrangement of Supply and Return connections readily lends itself to every installation requirement and permits installation in any position, tubes vertical or horizontal, or Units "laid flat" for vertical air flow.

Uniform steam or cold water distribution thru every tube is assured by proper orifice restriction at supply ends of tubes, this being accomplished without use of separate orifice rings, by machine-shaping the tube ends.

Complete descriptive and Engineering Data are presented in our new *Bulletin G32*, which Architects, Engineers and Contractors are urged to secure at once by request to Newark.

Design and Construction: The heat-transfer surface in AEROFIN is a plurality of seamless copper tubes about which is wound a helix of copper ribbon, crimped on its inner edge to permit winding and to afford maximum contact between tube and fin. The extended fin surface is applied and tinned while held in position, by highly developed automatic machines, being accurately crimped and spaced. The tinning of the tube and the extended surface makes them metallicly integral, affording maximum heat transmission and permanent effectiveness. The thickness and width (height) of the extended surface, the crimping and the pitch of the helix were determined by careful experiment, to afford maximum heat transfer, uniform air flow and minimum resistance thereto. The copper tubes and fins of AEROFIN transmit heat eight times as effectively as iron. So scientifically is AEROFIN designed that air is heated more in passage through a single row of its tubes, a travel of 1⅛ in., than in passage through an entire section of cast-iron surface, a travel of 9 in.

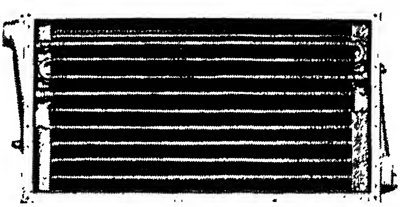


Fig. 2

Flexitube AEROFIN is constructed as briefly described above. In Universal AEROFIN and High Pressure AEROFIN the seamless tubes, with their extended fin surface, are continuous. (See Figs. 2 and 3.)

Flexitube AEROFIN, Universal AEROFIN and High Pressure AEROFIN are furnished as completely encased Units, ready for pipe and duct connections. The casings are built of pressed steel and are exceptionally strong and rigid, protecting the Unit from all the strains of pipe connections and expansion or contraction in service. The casings are flanged on both faces, top and bottom, and template punched for bolting together adjacent Units, or for duct connection.

Standard Casings: The casings of all AEROFIN Units (including the new Flexitube, which is interchangeable with former

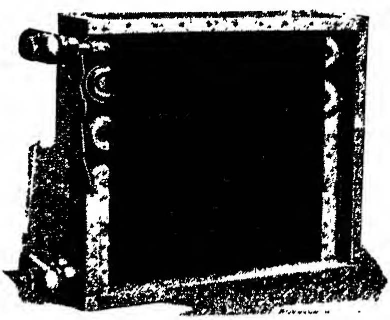


Fig. 3

Low Pressure), whether comprising one, two or three rows of tubes, are 29 in. wide (except 6-tube Universal AEROFIN which is 21$\frac{5}{16}$ in.) across tubes, from outer edge one flange to outer edge opposite flange, and 10 in. deep in direction of air flow. Length of casing is nominal tube length plus 8½ in.

Aerofin Sizes: Flexitube AEROFIN: Made in thirteen standard tube lengths, either one or two staggered rows of tubes per Unit, in 5 series (as explained in Bulletin). There are thus sixty-five standard Units available, a range which adequately meets all requirements.

Tubes are furnished of any length between 2 ft. and 6 ft., in increments of

6 in., and between 6 ft. and 10 ft. in increments of 1 ft.

Complete tables of sizes and capacities are shown in *Bulletin G32*, mailed gratis upon request. This bulletin also contains 19 proved Piping Diagrams in four colors.

Universal Aerofin: (Fig. 2). Available in one-row or two-row Units, 6-tubes or 9-tubes across face, in seventeen standard tube lengths (distance between end baffle plates, or between 180° bends) between 2 and 10 ft., inclusive, in increments of 6 in. Batteries of double width are easily assembled by setting Units end-to-end, leaving space for pipe connections.

Complete information is contained in our *Bulletin G32*, mailed gratis upon request.

High Pressure Aerofin: (Fig. 3). Made in five standard tube lengths (*i.e.*, length of straight section of tubes, between 180° bends) 2 ft.; 2 ft. 6 in.; 3 ft.; 3 ft. 6 in.; 4 ft.. either one, two or three staggered rows of tubes per Unit. There are thus fifteen standard units now available, meeting practically all requirements. Since the supply and drip connections of High Pressure AEROFIN are located on the same end (Figure 3) Units may be placed end-to-end, thus affording battery widths of twice the standard tube-lengths. Complete tables of sizes and capacities are shown in our *Bulletin G32*, mailed gratis upon request.

Narrow Width Aerofin: (Fig. 4), is to be used for Cooling only. The construction of Unit is the same as Flexitube AEROFIN except that tubes are straight and depth of casing is "5 instead of 10 in." This saves half the space required, in direction of air flow.

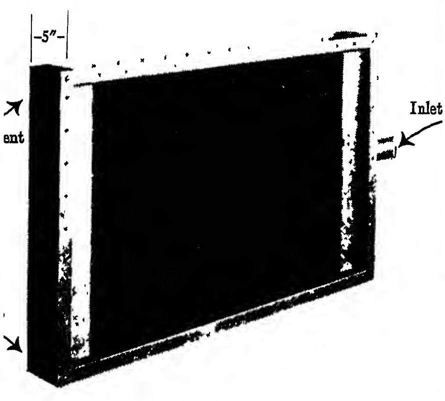


Fig. 4

Note that brass air-vent connection is installed at top of outlet end, outlet being shown, in illustration, at bottom of Unit, with inlet in center on opposite end.

Dimensions other than given above are the same as in Flexitube AEROFIN.

Aerofin Booster Units: (Fig. 5) for horizontal or vertical air flow. Six sizes 150 to 1624 c.f.m., 200 lbs. working steam pressure. Complete information upon request to Newark. Ask for *Bulletin G32*.

Steel Supporting Legs: Standard steel supporting legs, 18 or 24 in. high, template punched to same bolt hole centers as standard casing, are furnished when ordered. These legs may be attached quickly and obviate necessity of any other foundation.

Advantages: AEROFIN weighs but 9 to 16 per cent as much as equivalent cast-iron and occupies but 35 per cent of the space required by equivalent cast-iron (2-row units). Two men can easily carry any AEROFIN Unit. Expensive foundations are unnecessary, building re-enforcement is not required and the Units may readily be suspended from beams or roof trusses, or installed snugly in any out-of-the-way corner.

Ask Newark for *Bulletin G32* at once.

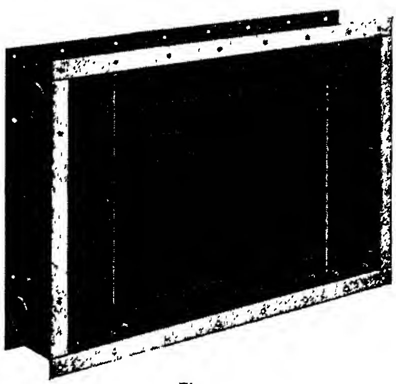


Fig. 5

Sale: AEROFIN is sold only by manufacturers of nationally advertised Fan System Apparatus. List upon request.

The G & O Manufacturing Company

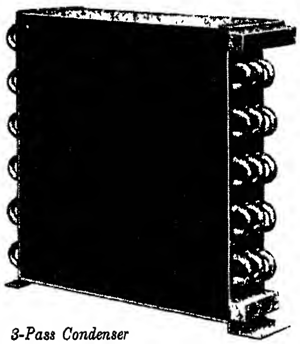
138 Winchester Avenue **New Haven, Connecticut**

G&O

INDIVIDUAL FIN TUBING

for

Unit Heaters (high and low pressure), Concealed Radiation, Refrigeration Condensers, Evaporators, Intercoolers, Aftercoolers, Cooling Units, and other Heat Transfer Products.



3-Pass Condenser

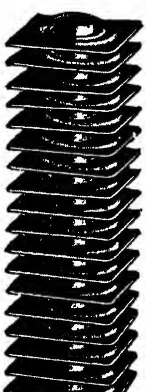
Individual G & O Fins—The use of individual fins results in high efficiency in heat transfer from primary tube surface to secondary fin surface, because all G & O fins have ample collars around the tube opening insuring liberal contact with tubing.

Any Size or Shape—Fins of any size or shape may be obtained giving any desired proportion of primary and secondary radiating surface.

Square Fins—A square fin has about 30% greater surface than a round fin of a diameter equal to a side of the square.

Individual Fins—Individual fins permit of any fin spacing: also, of using fins in groups at intervals along tubes. Fins placed at practically any angle on tubes.

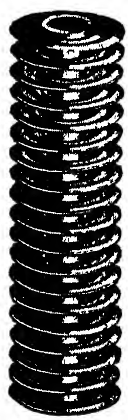
Various Shapes—G & O individual fin tubing is furnished in straight lengths: in U bends with short radii: in continuous return bend coils, or other shapes. Ends of tubes may be left free of solder for mechanical joints.



G & O Individual Fin Tube Data

Standard Sizes

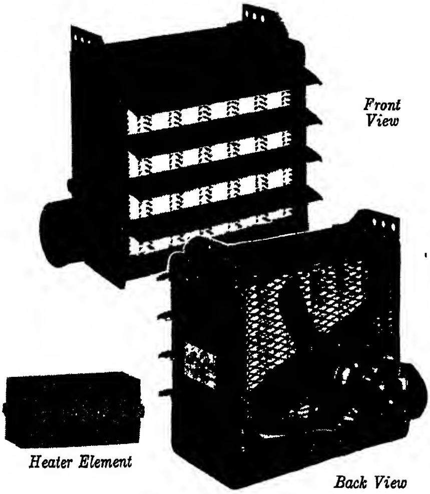
O.D. of Tube	Fin Size	Fin Spacing per Inch	Surface per Linear Foot
3/8″	3/4″ sq.	6	0.59 sq. ft.
3/8″	7/8″ sq.	6	0.81 sq. ft.
3/8″	7/8″ r'd.	6	0.64 sq. ft.
3/4″	1½″ r'd.	6	1.61 sq. ft.
3/4″	1 5/8″ sq.	6	2.50 sq. ft
1″	2 1/8″ sq.	6	4.00 sq. ft.
1¼″	2½″ r'd	5	3.55 sq. ft.



American Foundry Equipment Company

Mishawaka, Indiana, U. S. A.

Manufacturers of Industrial and Domestic All-Electric Unit Heaters



Front View

Heater Element

Back View

American All-Electric Unit Heater

Electromode "Built-In" All-Electric Heater

Thermofan-Portable All-Electric Heater

Experience of hundreds of users and many operation tests by power and public service interests, insurance companies, United States Government and other authorative institutions have proved that the American All-Electric Heaters are practical for hundreds of varied applications.

They incorporate the unit heater principle in which forced air circulation provides uniform temperature. They are clean, convenient, instantaneous, efficient, fully automatic and portable. The low initial cost and minimum installation expense enables economical heating for many applications.

The heart of the American Heater is the heating unit. This consists of a helicoil, sheathed wire type (Calrod) of resistance heater element cast integral with an aluminum alloy fin type grid. The heat transformed from electric energy is quickly conducted through the finned area of the heating unit and is carried off by the forced air circulation. The operating temperature of the unit is comparatively low and with the special cut-out switch, absolute safety is assured.

In addition to the industrial type, the American Heaters are made in domestic models. The built-in type and the portable model are both ideal for home use where occasional heat is required.

Write us for complete information on the American Line of heaters

RATINGS

INDUSTRIAL HEATER

Model No.	K.W.	C.F.M.	B.T.U.	E.D.R.
83–12	1.2	75	4,098	17.1
83–15	1.5	95	5,122	21.3
83–18	1.8	95	6,147	25.7
84–20	2.0	150	6,830	28.5
84–24	2.4	150	8,196	34.2
84–30	3.0	210	10,245	42.7
84–40	4.0	210	13,660	57.0
12–6	6.0	375	20,490	85.4
12–9	9.0	560	30,735	128.0
12–12	12.0	625	40,980	170.8
18–135	13.5	1055	46,102	192.1
18–1575	15.75	1055	53,786	224.1
18–18	18.0	1055	61,470	256.1
18–27	27.0	1410	92,205	384.2
24–36	36.0	2815	122,940	511.8
24–42	42.0	2815	143,430	597.6
24–48	48.0	3130	163,920	683.0
24–60	60.0	3130	204,900	853.8

ELECTROMODE

Model	K.W.	C.F.M.	B.T.U.	E.D.R.
A1200	1.2	110	4,098	17.1
A1600	1.6	110	5,464	22.8
A2000	2.0	110	6,830	28.5
B3000	3.0	180	10,245	42.7

All models supplied either for 2 in. by 4 in. or 2 in. by 6 in. stud mountings.

THERMOFAN

Model	Switch	Current
P.	Plug in Type	110V-A.C.
1	Heat Switch in Base	110V-A.C.
2	Heat and Fan Switches in Base	110V-A.C.

All models either 1000—1200—or 1600 Watts.

Thermal Units Manufacturing Company

Pershing Road and Loomis Street, **Chicago, Ill.**

NEW YORK OFFICE: 30 Church Street — REPRESENTATIVES IN OTHER PRINCIPAL CITIES

PRODUCTS—Unit Heaters, Unit Coolers, Air Conditioning Units and Apparatus, Dehumidifyers.

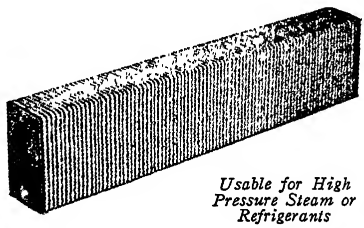
Also Blowers, pressure or volume; Coolers for gas, oil or water; Dryers; Evaporators; Fans, ventilating or exhaust; Air Heaters for dryers; etc.; Heating and Ventilating Units combined; Concealed Radiation; Refrigerating and Ice Making Machinery and Plants.

The Thermal Unit Integrally Cast, One-Piece Aluminum Alloy Element

No Joints, Welds, Brazed or Soldered Connections.

Freezeproof, Corrosion and Acid Resisting.

Leakproof—Indefinite Life Without Servicing.



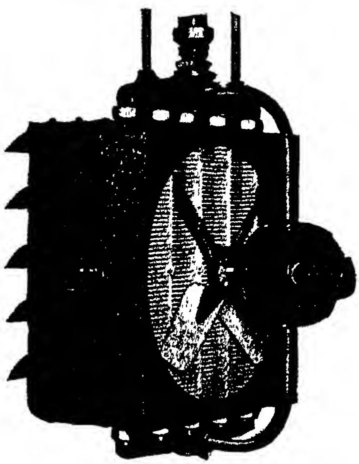
Usable for High Pressure Steam or Refrigerants

Neutral to All Refrigerants.

High Heat Transfer.

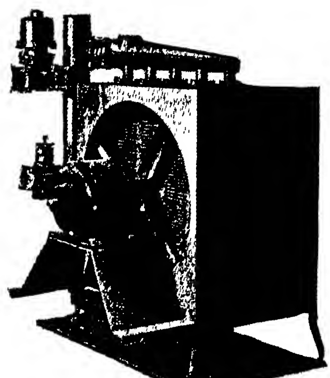
Tested to 450 lbs. Hydrostatic Pressure.

Lowest Air Resistance.



Note the Trim Assembly and Sturdy Construction of the Units

Troubleproof Service and Long Life



Thermal Unit Heaters

Model No.	R.p.m.	Steam Pressure, Lbs.	B.t.u. at 60°	C.f.m.	Final Temperature	Motor H.P.
		5	47,600	1101	100	
12	1750	100	79,100	1101	127	1/10
		5	83,600	1958	100	
16	1150	100	140,000	1958	127	⅛
		5	132,000	3060	100	
20	1150	100	219,900	3060	127	1/6
		5	188,000	4406	100	
24	1150	100	314,800	4406	127	¼
		5	274,100	6426	100	
30	1150	100	459,200	6426	127	½

Thermal Unit Coolers

Rust resisting construction throughout, and manufactured in five sizes, with capacities of ½ ton to 8 tons per unit depending on refrigerant temperature and final temperature desired.

Suitable for cold storage or any process or product requiring cooling or dehumidifying.

For use with Ammonia, Methyl Chloride, Sulphur Dioxide, Freon, Brine or cold water.

Defrosted and controlled automatically.

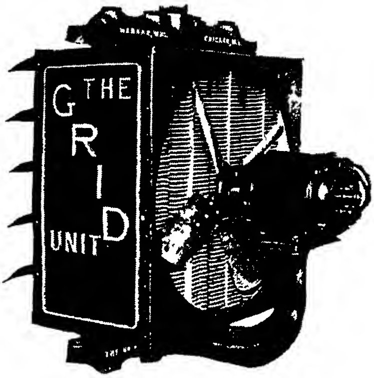
Complete Catalogues, Information and Engineering Data furnished on request

The Unit Heater and Cooler Co.

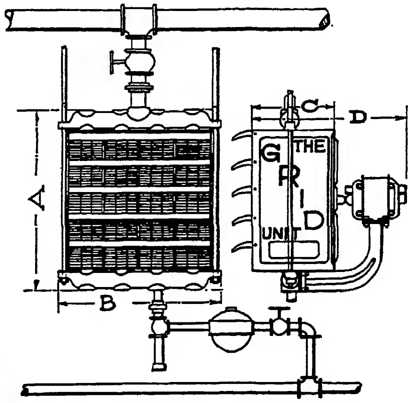
Wausau, Wisconsin

Offices in Principal Cities

MANUFACTURERS OF THE GRID UNIT



Sturdy—Permanent
THE GRID UNIT HEATER
(A product of the D. J. Murray Mfg. Co.)



Typical Installation of Grid Unit Heater

GRID UNIT HEATER DATA

Delivers More Warm Air to Floor and Working Zone

Model No.	Dimensions				Face Area Sq. Ft.	Motor		Volume at Fan	Capacities 5 Lb. Press. 60° Air		Approx. Shipping Weight	Pipe Sizes	
	A	B	C	D*		Hp.	R.P.M.		B.T.U.	Final Temp.		Supply	Outlet
1200	15	12½	11¼	17½	1.04	1/20 1/30	1700 1150	711 515	46000 34900	119 122	120	1¼″	1¼″
1500	22	18	11½	20	1.67	1/10 1/20	1750 1150	1450 1015	77500 65500	109 118	210	1¼″	1¼″
1520	27	18	11½	20	2.2	1/8 1/20	1750 1150	1700 1288	104000 83500	113 118	250	1¼″	1¼″
2000	27	23¼	11½	21½	2.8	1/6 1/10	1150 850	2500 1835	148000 113000	114 117	320	2″	1¼″
2025	32	23¼	11½	21½	3.6	1/6 1/10	1150 850	2875 2380	177000 158500	115 120	370	2″	1¼″
2500	32	28½	11½	28	4.5	1/2 1/4	1150 850	4200 3375	225000 197000	108 113	440	2″	1¼″
2530	36	28½	11½	28	5.3	1/2 1/4	1150 850	4650 3350	282000 226000	115 121	500	2″	1¼″
3000	38	33	13¾	29	6.5	1 1/2 1/2	1150 850	8100 6350	394000 341000	104 109	725	2½″	1¼″

*Varies with type of motor.

Long Life—Efficient Service—Good for 250 lbs. working pressure.
Steam Chamber—Cast High Test Iron.
Fin Surface—Aluminum alloy cast metal to metal contact with steam chamber.
Manifolds or Headers—Cast high test iron.

Lower outlet temperatures and greater volume delivers warm air to working zone of room, thus eliminating heat losses at ceiling and avoiding stratification.

Grid Unit Heaters—Reduce fuel cost and maintenance expense—no damage possible to **Grid** due to electrolytic action as it cannot take place in **Grid Units.**

Cooling and Refrigerating—**Grid Unit Cooler** ideal for cooling, using cold water, sodium or calcium brine or direct expansion ammonia.

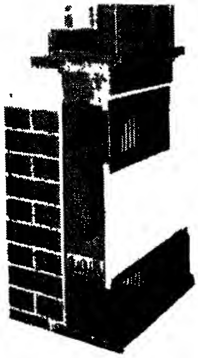
Write for further information and catalog.

Young Radiator Company

Unit Heaters, Convection Heaters, Blast Heaters, Heat Transfer Surfaces, Unit Coolers

Representatives in all Principal Cities

Home Office: **Racine, Wis.**

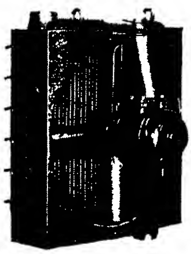


Sectional view of plastered-in enclosure

Young *Streamaire* copper convectors are furnished with plastered-in type enclosures, metal front recess cabinets, wall-hung cabinets and free-standing cabinets. These present a pleasing appearance, are solidly built, extremely compact and highly efficient in operation. The patented construction of Young *Streamaire* convectors insures positive satisfaction when operating on vapor, vacuum, or one-pipe steam systems or on gravity or forced circulation hot water systems.

Unit Heaters

Type "SH" for Steam or Hot Water—to heat garages, stores, industrial, and similar type buildings. Made in 17 different models, equipped with constant or multi-speed motors. Recirculating ducts, wall boxes, and heat diffusing nozzles can be furnished with all models; also air filters when desired. DeLuxe models have chromium plated housings.



Unit Heater

Type "VH"

Slow fan speed and low power cost assures quietness and economy in this model. Blower type fans circulate a large volume of heated air. An ideal unit to operate in conjunction with air ducts.

Blast Heaters and Commercial Units

Blast Heater

Made with heavy seamless copper tubes and copper fins (extended heating surface). Ideal for use with forced air heating and ventilating systems. For use with high and low steam pressures or with hot water. Designed so that all internal strains due to expansion and contraction are eliminated. Write for catalogue giving sizes and capacities.

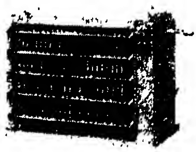
Recess Cabinets Free-Standing Cabinets

Are availabe in various sizes and capacities to fill every heating need. They offer a complete compact enclosure for the heating element.

Free Standing Cabinet

Unit Coolers

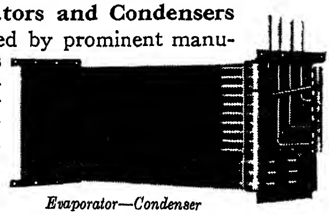
Types "XC" and "SC"—Made with two types of coils, one for use with ice water or brine as the cooling medium, and one for direct expansion use in connection with refrigerating machines using refrigerants such as sulphur dioxide, methyl chloride, F 12, ammonia, etc., where they are to be used to maintain room temperatures above the freezing point.



Unit Cooler

Evaporators and Condensers

Selected by prominent manufacturers of refrigerating and air conditioning equipment for special, heavy service and exceptional applications. A great variety of tool, die and assembly equipment is available to manufacture these units to meet almost any specification.



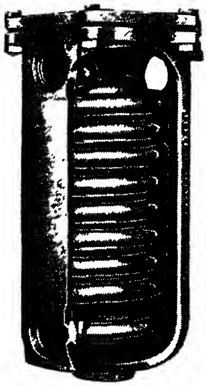
Evaporator—Condenser

Bell & Gossett Company

3000 Wallace Street **Chicago, Ill.**

Steam and Vapor

B & G Indirect Water Heaters for Steam and Vapor Systems

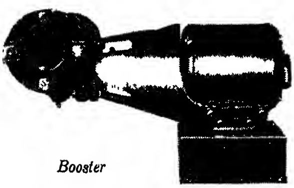


Description	No.	Capacity Gallons	Max. Length Inches	Max. Width Inches	Shell Openings Inches	Coil Openings Inches	Shipping Weights Pounds
SINGLE COIL For residences of all sizes. Duplex apartments and small buildings.	30	30	11 3/4	5 1/16	1	3/4	12
	40	40	12 3/4	5 1/16	1	3/4	13
	52	52	14 3/4	7 1/8	1	3/4	15
	66	66	15 1/8	7 1/8	1 1/2	1	29
	82	82	18 3/4	7 1/8	1 1/2	1	39
	100	100	20 3/4	7 1/8	1 1/2	1	40
	120	120	22 3/4	7 1/8	1 1/2	1	42
	144	144	25 1/4	7 1/8	1 1/2	1	46
DOUBLE COIL For larger apartments, garages, medium sized factories and office buildings.	160	160	12	11 1/2	2	1 1/2	53
	192	192	14 3/4	11 1/2	2	1 1/2	59
	300	300	19 5/16	11 1/2	2	1 1/2	78
	400	400	23 3/8	11 1/2	2	1 1/2	86
TRIPLE COIL For heavier requirements.	600	600	21 3/8	15 3/4	3	2 1/2	230
	800	800	25 3/8	15 3/4	3	2 1/2	250
	1000	1000	29 1/4	15 3/4	3	2 1/2	290
DOUBLE TRIPLE COILS	1200	1200	23 3/4	29	4	3 1/2	568
	1600	1600	27 1/2	29	4	3 1/2	642
	2000	2000	30 3/4	29	4	3 1/2	716

Above ratings based on 100 degree rise in three hours with boiler water temperature of 180 degrees or more.

HOT WATER—B & G Triple Duty Systems for Hot Water Systems

Flow Control



Booster

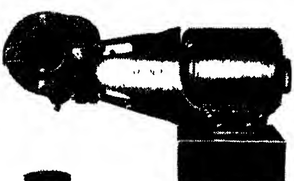
B & G Heater

These three units comprise the Triple Duty System

No.	Recommended Tank Sizes	Recommended Number of Baths
19	30 to 42 Gallons	1
20	42 to 52 Gallons	2
21	52 to 66 Gallons	3
22	66 to 120 Gallons	4
23	120 to 180 Gallons	5
24	180 to 300 Gallons	6

Important—This system will furnish year round domestic hot water supply from any oil, gas, stoker or hand fired **hot water heating boiler.**

B & G BOOSTER—Automatic Electrically Operated Circulator with Positive Centrifugal Pump Action



No.	Type	Flange Size	Capacity	Motor	Style
1½-30	Industrial	1½ Inches	30 Gallons per Min.	1/8 H.P.	Repulsion Induction
1½-30	Residential	1½ Inches	30 Gallons per Min.	1/8 H.P.	Capacitor
2-50	Industrial	2 Inches	50 Gallons per Min.	1/6 H.P.	Repulsion Induction
2-50	Residential	2 Inches	50 Gallons per Min.	1/6 H.P.	Capacitor
3-100	Residential	3 Inches	100 Gallons per Min.	1/4 H.P.	Capacitor

B & G THERMO-CHECK

Controls automatically the temperature of hot water storage tanks heated by indirect heaters. A simple device that requires no attention and also eliminates lime and sediment formation in the heater coils. Sizes 1½″ and 2″.

B & G Motorized Valves—Sizes 2″ to 6″ in both Straightway and Angle Patterns

101 Park Ave.

McDERMOTT WATER HEATERS, INC.

New York City

SOLD BY JOBBERS AND BOILER MANUFACTURERS

McDERMOTT WATER HEATERS, INC., installed the first submerged water heater in New York City. Today McDermott Heaters are installed in buildings supplying hot water to over 48,000 apartments in the New York metropolitan area.

These heaters may be used as oil preheaters and converters for any usual adaptation.

We manufacture water heaters exclusively. We believe we can show them to be superior in quality and performance. We are prepared to meet price competition anywhere.

Complete information on request.

Table 2—McDermott In-Boiler Heaters
(Used with Storage Tanks)

Cat. Size No.	Lgth., Ft.	Cap. Gals.*	Tank Con., In.	Boiler Opening, In.	List Price
V4103	3	150	1¼	1¼x4¾	$30.00
V4106	6	245	1¼	1¼x4¾	50.00
V4108	8	350	1¼	1¼x4¾	66.67
M4210	10	550	2	3⅜x3½	100.00
M4212	12	800	2	3⅜x3½	116.67

*Ratings based on 100° temperature rise in three hours, 180° F. boiler water in summer with supply at 75° F., and 212° F. boiler water in winter with supply at 45° F.
Special lengths take price of next larger size.

Table 1—McDermott Exterior Heaters and McDermott In-Tank Heaters

Cat. size No.	Capacity, Gals.: *With steam at 1-lb. pressure	Capacity, Gals.: ‖Using water from steam heating boiler as the heating medium	Capacity, Gals.: †Using 180° F. water as the heating medium	Pipe Connections, In.: Coil: Steam	Pipe Connections, In.: Coil: Water	Pipe Connections, In.: Shell: Steam	Pipe Connections, In.: Shell: Water	Dimensions in Inches: L: Over-all length	Dimensions in Inches: D: Over-all Diameter	Square feet of copper	Weight with shell (crated), lbs.	‡List Price: Complete with shell
52	180	110	77	2	2	1	2	30	8	3.14	78	$24.00
53	280	165	116	2	2	1	2	42	8	4.71	98	40.00
54	370	220	154	2	2	1¼	2	54	8	6.28	117	50.00
55	465	275	193	2	2	1¼	2½	66	8	7.85	136	60.00
56	560	330	231	2	2	1½	2½	78	8	9.42	155	65.00
57	650	385	270	2	2	1½	2½	90	8	10.99	174	72.00
58	740	440	308	2	2	2	3	102	8	12.56	193	80.00
59	840	495	347	2	2	2	3	114	8	14.13	212	90.00
510	920	550	385	2	2	2	3	126	8	15.70	230	100.00
103	1100	660	462	2	4	2	4	56	16	18.84	405	105.00
104	1480	880	616	2	4	2	4	68	16	25.12	459	135.00
105	1840	1100	770	2	4	2	4	80	16	31.40	513	216.00
106	2220	1320	925	2½	5	2½	5	92	16	37.68	567	258.00
107	2580	1540	1078	2½	5	2½	5	104	16	43.96	621	300.00
108	2950	1760	1232	2½	5	2½	5	116	16	50.24	675	350.00
109	3320	1980	1386	3	6	3	6	128	16	56.52	729	400.00
1010	3690	2200	1540	4	6	4	6	140	16	68.80	783	450.00
125	2775	1650	1155	2½	5	2½	5	84	19	47.10	628	415.00
126	3330	1980	1386	3	5	3	5	96	19	56.52	697	460.00
127	3885	2310	1617	4	6	4	6	108	19	65.94	766	505.00
128	4440	2640	1848	4	6	4	6	120	19	75.36	835	550.00
129	4995	2970	2079	5	8	5	8	132	19	84.78	904	595.00
1210	5550	3300	2310	5	8	5	8	144	19	94.20	973	640.00
83	1330	792	555	2	4	2	4	54	13½	18.90	303	88.00
84	1774	1056	740	2	4	2	4	66	13½	25.2	357	188.00
85	2220	§1320	§925	2½	5	2½	5	78	13½	31.5	411	238.00
86	2662	§1584	§1110	2½	5	2½	5	90	13½	37.8	465	288.00
87	3106	§1848	§1295	3	6	3	6	102	13½	44.1	519	338.00
88	3550	§1480	§1480	4	6	4	6	114	13½	50.4	573	386.00

*Gallons raised 130° F. in 3 hrs. ‖Gallons raised 100° F. in 3 hrs. Based upon 180° F. boiler water in summer with supply at 75° F. and 212° F. boiler water in winter with supply at 45° F. †Gallons raised 100° F. in 3 hrs. Based on supply at 45° F. §Use with circulating pump if heating medium is water. ‡Write for discounts.

This McDermott Heater was manufactured by us for the U. S. Government

Table 3—McDermott Tankless-Instantaneous Water Heaters Using Water or Steam as the Heating Medium—Installed in Boilers or in Self-contained Shells

Size No.	V3	V4	V5	V6	V7	V8	M10	M12
In Steam Boiler — Gals. raised 130° F. hourly — 40 lbs. velocity pressure*	650	800	1000	3500	5500	7300	14000	17000
In Water Boiler — Gals. raised 100° F. hourly — 40 lbs. velocity pressure*	455	560	700	2450	3850	5110	9800	11900
Square feet of surface	10.5	13.65	16.80	19.95	23.1	26.25	50.24	54.95
Boiler opening, in.	1½x15.5	1½x15.5	1½x15.5	1½x15.5	1½x15.5	1½x15.5	3.75x12	3.75x12
Shell dimensions, in	10x	10x	10x	10x	10x	10x	14x	14x
If self-contained, ft.	3	4	5	6	7	8	10	12
L—length, ft.	3	4	5	6	7	8	10	12
W—width, in.	15.25	15.25	15.25	15.25	15.25	15.25	11.5	11.5
Weight with shell, crated	198	248	298	348	398	448	810	880
Weight without shell, crated	45	52	59	66	73	80	170	180
No. of units in Table 2 of equal capacity§	3V4103	3V4104	3V4105	3V4106	3V4107	3V4108	3M4210	3M4212
List price with shell	$120.00	$150.00	$165.00	$190.00	$215.00	$240.00	$370.00	$410.00
List price without shell	$90.00	$110.00	$125.00	$150.00	$175.00	$200.00	$300.00	$350.00

*Maximum discharge temperature 30° F. less than boiler water temperature. §Smaller boiler openings and greater efficiency in maintaining primary heat on transfer surface result from *non-grouped coils.* They are connected in series by piping contractors.

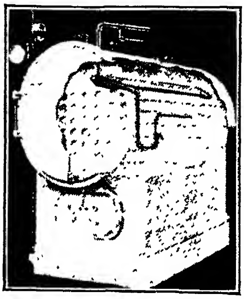
Parkinson Heater Corporation

Parkinson Tankless Submerged Water Heaters, Tempering Valves, etc.

11 West 42nd St., **New York City** — 4105 N. Damen Ave., **Chicago, Ill.**

Local Distributors in Principal Cities. Consult phone book or write

PARKINSON
TANKLESS SUBMERGED
WATER HEATER



One of the typical installations Parkinson in a fire tube boiler

Officially approved by the Investigatin Committee of Architects and Engineers.

PRODUCTS—Parkinson Tankless Submerged Water Heaters for domestic water supply, designed to eliminate storage tanks, insulation, separate heater, extra flue, fittings, piping and hook up at a very definite savings in fuel costs. Holby Tempering Valves (write for details) for use with water heaters to increase gallon capacity and assure uniform temperature within 5 degrees and permit continuous circulation.

Free Service—Without obligation, our Engineering Department will submit recommendations, layouts, or advice on domestic hot water and piping requirements.

Send for list of users, actual case studies and circulars.

Guarantee—Parkinson Heaters are guaranteed to produce under normal conditions a saving of from 30% to 70% of fuel costs. All parts tested under 250 lbs. pressure.

How to Figure Heater Required

Tankless water heating cannot be figured by the customary method as used for tanks. Therefore the following method using the "Demand Unit" is used. This unit has been developed from the result of thousands of installations over six years.

Calculation of Demand Units	Units
Kitchen and bathroom, normal	10
Extra apartment bath, private	3
Extra apartment bath, maid	4
Kitchen	5
Cupboard kitchen	2
Bathtubs, normal	5
Separate shower	5
Shower in tub	1
Lavatories	2
Club or institution showers	25
Sinks (slop)	3
Restaurant (hand washing)	1 per seat
Wash tubs, apt. laundry	5
All other loads: the detail of equipment should be furnished.	

Demand Units	Comparative Capacity Standard Estimated Tank Size, Gals.	Installation No. Water Pressure Up to 40 Lbs.	40 to 80 Lbs.	80 to 120 Lbs.
10	50	1	1	3
20	90	3	3	3
30	120	3	3	6
60	240	6	6	12
120	480	12	12	12
160	600	12	18	20
220	840	18	18	20
280	1080	20	30	30
360	1400	30	30	50
420	1680	30	50	50
480	1900	50	50	70
540	2150	50	50	70
600	2400	50	70	70
660	2700	50	70	224
720	2900	70	224	224
840	3100	70	224	224
1000	5400	250	250	270
1500	6400	270	270	270
2000	7000	270	290	290
2500	9000	290	290	290

NOTE—*Put the letter C after installation number if it is a cast-iron boiler.*

Piping details furnished per request. To estimate the cost of piping same as hooking to tank.

Load on Boiler

Steelflue type boiler—Due to increased efficiencies set up in the boiler it is safe to install PARKINSONS in boilers without additional radiation provided the boiler is ample for the heating load. For new boilers a factor of safety of 2 sq. ft. per demand unit may be figured.

Cast-iron boilers require 4 sq. ft. radiation for each demand unit.

Water Temperature

Line temperatures range from 135° to 165°, but may be controlled within 15°. For hotter water special heater is required. *Holby Tempering* Regulators are supplied for more uniform requirements built special for Parkinson Heater.

A Size for Every Purpose

The 3rd largest and 3rd smallest size Parkinson Heaters compared to a normal man.

Construction—The Parkinson is protected by Pat. No. 1,834,070 and operates on an exclusive principle not to be confused with other heating coils. Supply tube and return tubes are sized to produce a rapid and constant flow of hot water. Heater is constructed of virtually pure copper, consisting of heavy wall tubes and cast headers, brazed together with copper flux at high temperature.

Installation—Heating unit is submerged in boiler water just below water line of practically any low pressure boiler. Simple to install. Does not multilate the boiler as only one incision is required. Unit takes up no usable space in cellar. Operates on any fuel, automatic or hand fired.

Chase Brass & Copper Co.

INCORPORATED

Waterbury **Connecticut**

For data on Chase Copper Radiators see page 676

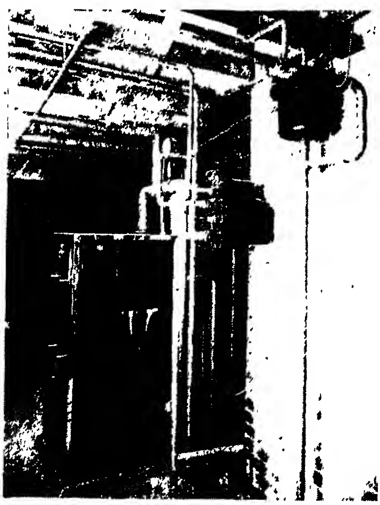
CHASE

Rustless Heating Lines of Chase Copper Tube and Chase Sweat Fittings

Here are a few reasons why copper tube should be used for heating lines: **1.** Copper does not rust, the water, vapor, or steam inside the lines is kept clean. **2.** Delicately adjusted thermostatic valves and traps are not rust-clogged out of commission. **3.** Rustless heating pipes also eliminate the need for dirt pockets in steam mains. **4.** Copper is inexpensive and as satisfactory for giving service as any material known.

TEMPER—Both hard and soft tubing are available for heating lines. In remodelling work, and in replacing old heating systems the soft copper tubing will be found particularly useful. It can be worked down between walls and around corners, long 60 ft. coils eliminate many useless connections. For all soft copper tubing we recommend the extra heavy gage, (known by United States Government specifications as "Type K"). For the average new installation we recommend the light gage copper tube ("Type M").

PRESSURE—Chase copper tube is adaptable for all low pressure heating systems. While we do not recommend the use of copper tubing with any steam system having more than 30 lbs. pressure, the factor of safety over such pressure is very great. The bursting pressure, for example, of ¾ in. Chase copper tubing and sweat fittings when connected together is 3,050 lbs. per square inch.



RETURN LINES—While we recommend copper tubing for all heating lines, it is in the return lines that the greatest amount of rusting takes place. Architects and Engineers have learned by experience that these are the lines that are apt to rust and in larger installations need replacement, after a very short period of service.

CHASE

This is a typical vapor installation. Notice the neatness of the copper tube and sweat fitting heating lines. Also the absence of union connections.

CHASE COPPER TUBE FOR HEATING LINES

CHASE

Copper tube and sweat fittings are used for the run-outs from main to risers. This minimizes resistance to flow of steam.

Steam Carrying Capacities

The steam carrying capacity of copper tubing is greater than that of iron pipe of the same nominal size. For example, in a steam main, 2 in. copper tubing is capable of taking care of a load of 444 sq. ft. of radiation as compared with 386 sq. ft. for 2 in. iron pipe. This is an increase of 15.1 per cent.

For equal steam pressure drop the capacity of copper tubing is on the average 10 per cent greater than for the same nominal size of iron pipe. Because of this it is frequently possible to use tubing of smaller size.

Pipe Covering

Copper Water Tubing uses the same standard pipe coverings as other heating pipes, but in most cases requires one size smaller covering. This is a saving in the cost of covering.

Costs

The slight extra cost of Chase copper tubing and sweat fittings is a very small amount in dollars to pay for the advantages of a copper tube installation.

SIZES AND WEIGHTS

Nominal Size	Outside Diameter	Inside Diameter	Wall Thickness	Pound per Lineal Foot
½ Inch	0.625 Inch	0.569 Inch	0.028 Inch	0.203 Lbs.
¾ "	0.875 "	0.811 "	0.032 "	0.328 "
1 "	1.125 "	1.055 "	0.035 "	0.464 "
1¼ "	1.375 "	1.291 "	0.042 "	0.681 "
1½ "	1.625 "	1.527 "	0.049 "	0.940 "
2 "	2.125 "	2.009 "	0.058 "	1.46 "
2½ "	2.625 "	2.495 "	0.065 "	2.03 "
3 "	3.125 "	2.981 "	0.072 "	2.68 "
3½ "	3.625 "	3.459 "	0.083 "	3.58 "
4 "	4.125 "	3.935 "	0.095 "	4.66 "

John J. Nesbitt, Inc.

AND

Buckeye Blower Company

Manufacturers of Heating, Ventilating and Air Conditioning Equipment

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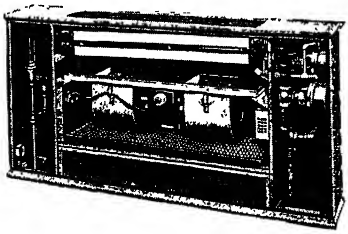
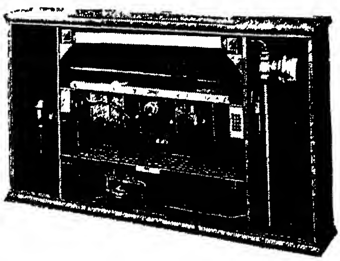
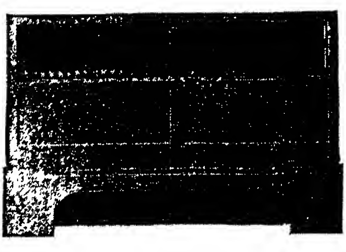
Sales and Service on Nesbitt Universal School Room Unit Ventilators through Offices of American Blower Corp.



Universal Type "D" and Buckeye Series 900 Double Radiator Duo Luxe Heating and Ventilating Units—Tomorrow's Heating and Ventilating Unit Today—insures healthful, comfortable rooms without drafts by syncretizing room temperature with air-stream temperature. Air capacities from 450 to 1560 c.f.m. As today's most economical unit, this unit delivers a constant amount of outdoor air heated to the proper temperature. As fast as State codes are changed this unit can be adjusted to mix indoor and outdoor air in any desired proportion, thus it is also tomorrow's most economical unit. Ask for Universal publication No. 217—Buckeye publication No. 133.

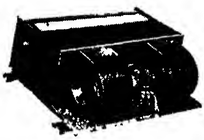


Universal Type "O" and Buckeye Series 400 Heating and Ventilating Unit—Always delivers outdoor air to occupied rooms in percentages governed by both indoor and outdoor temperatures. Heat required for ventilation is only that necessary to raise the air-stream temperature from 60° to 70°. The operation of this unit is exceptionally economical—syncretizes air-stream and room temperatures to perfect harmony, thus removing the cause of cold drafts and overheating. Air capacities 450 to 1560 c.f.m. Ask for Nesbitt publication No. 218 or Buckeye publication No. 138.



Nesbitt Concealed and Cabinet Radiators—A rugged copper multifin, copper tube radiator supplied with or without metal enclosures. Built in a variety of styles and sizes to meet the most exacting conditions. Complete information in Nesbitt publication No. 224.

Nesbitt Type "HC" Air Conditioner heats and humidifies or cools and dehumidifies—Mixes room air and outdoor air in the proportion desired—Filters all air delivered and distributes it evenly throughout the room. The Nesbitt Conditioner can be used in one or in all of the offices in a single building, store, restaurant, hotel or home—in new building or old. Made in a variety of sizes, having a wide range of performance. Ask for "Personal Weather"—Nesbitt publication No. 221.



Buckeye "E" Thermovent and Universal Type "E" Heating and Ventilating Unit for auditoriums, churches, etc., where quiet operation is essential in the delivery of a large quantity of air. Made in a variety of types to handle all outdoor air or all indoor air or a mixture of indoor and outdoor air. Capacities from 2,000 to 6,000 c.f.m. Ask for Buckeye publication No. 137 or Universal publication No. 220.

The Buckeye Giant Unit Heater—A blower, draw-through type Unit Heater for the economical heating of large areas in industrial plants, garages, airplane hangars. Made in a variety of types and sizes of capacities from 1,880 c.f.m., 125,500 B.t.u. to 20,000 c.f.m. and over 1,000,000 B.t.u. capacity. Equipped with Thermadjust by-pass feature. Ask for Buckeye publication No. 136.

Buckeye Unit Heater—A rugged, compact, suspended type, disc fan unit heater built in capacities up to 313,500 B.t.u. (2 pounds—60°) for steam pressures up to 150 pounds with single or multispeed motors. Ask for Buckeye publication No. 140.

All Buckeye Unit Heaters are tested and rated in accordance with the Code of the American Society of Heating and Ventilating Engineers and the Industrial Unit Heater Association.

Nesbitt and Buckeye manufacture a transfer surface for blast heating or cooling. This surface is regularly applied for cooling as well as heating in both the Giant and disc fan type unit heaters. Ask for Nesbitt publication No. 222 covering the Nesbitt Coolofan, a disc fan Unit Cooler.

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

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PRODUCTS AND SERVICES—

Complete Service on materials to Specification on Power Plant Piping, Industrial Piping, and Industrial Heating Systems; Fabricated Piping including Pipe Cutting and Threading, Pipe Bends, Welded Headers, Welded and Welding Fittings, Lap Joints and the Grinnell Triple XXX line of products for Super Power.

Grinnell Equiflo Valves for forced hot water heating systems; Grinnell Adjustable Pipe Hangers and Supports; Grinnell Cast Iron and Malleable Iron Pipe Fittings; Grinnell Malleable Iron Unions; Grinnell Welding Fittings; Grinnell Thermoliers (Unit Heaters); Grinnell Unit Coolers (for refrigerating service); Thermoflex Traps and Heating Specialties.

Also Humidifying Systems; Constant Level Size Circulating Systems; Piping for acids and other special materials.

Malleable Iron, Brass, Bronze and other Castings; Brass, Cast Iron, Wrought Iron and Steel Pipe; Seamless Steel Tubing in Iron Pipe Sizes.

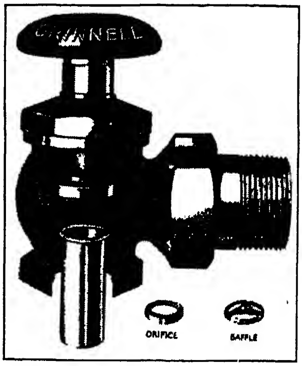
Valves: Check, Globe, Pressure Reducing and Regulating, Quick Opening, Safety and Y.

Automatic Sprinkler Systems; Stand Pipes; Underground Supply Mains; Hydrants; Fire Pumps; Pressure and Gravity Tanks.

Grinnell "Junior" Automatic Sprinkler Systems for Basements and other hazardous areas of Dwellings, Small Apartment Buildings, Schools, Churches, Stores, etc.

Grinnell Equiflo Valves

For Forced Hot Water Heating



Equiflo Valve

The designing of forced circulation hot water heating systems is so simplified by the Grinnell Equiflo Valve that they can be laid out and installed as easily as vapor or steam systems. This valve consists of a regular type packless radiator valve with a cartridge or tube made up of a series of orifices and baffles capable of setting up any required frictional resistance. This method of establishing any desired resistance does away with elaborate calculation of pipe sizes. Grinnell guarantees perfectly balanced circulation to each and every radiator where these valves are installed throughout the system.

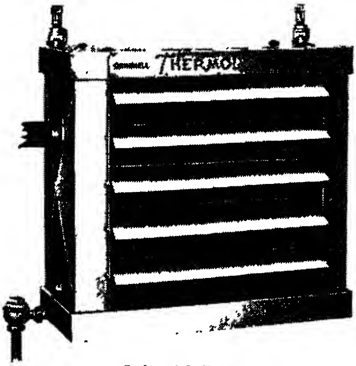
Equiflo Data Book sent to interested parties.

For Data on Thermoflex Traps and Heating Specialties, see page 745

Grinnell Thermolier

(Patented)

Industrial and Factory Types—125 Lbs. W.S.P.

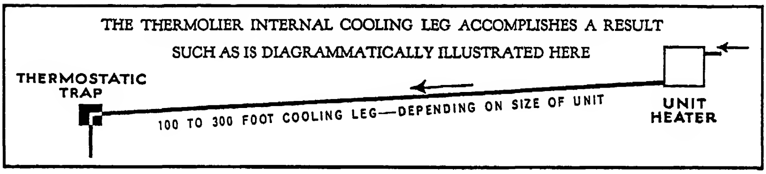


Industrial Type

Thermolier, the Grinnell development in "unit heaters," is a ruggedly built unit whose efficiency and dependability have been proved by actual performance in field service. Thousands of them are installed in industrial buildings and commercial structures of all types of occupancy.

Thermolier has 14 points of superiority, the most outstanding of which is the internal cooling leg built right into the unit, an exclusive Thermolier feature. See drawing below.

Radiation is from brass-finned seamless copper U-tubes rolled into a cast iron tube sheet. No solder is used for strengthening joints and there are no flat horizontal surfaces to catch dirt.



Units may be controlled manually or automatically, singly or in groups. Installation and piping are extremely simple and inexpensive, hence the unit may be moved from one location to another at small cost if found desirable on account of changes in building or occupancy. Furnished in sizes as listed.

Thermoliers provide maximum distribution of heat without objectionable drafts.

Specifications

Fan—Grinnell special of rugged construction. **Motor**—heavy duty, oversize, enclosed, moisture-proof. **Housing**—Copper on Industrial Type with rubbed lacquer finish; steel on Factory Type finished in gray duco. **Frame**—Heavy pressed steel, providing rugged support for motor and fan. **Special Features**—Adjustable swivel hanger rod couplings; louvers rigid, but easily adjustable: integral cooling leg insuring perfect drainage through one ½-in. trap on Models 100 and 200, ¾-in. on Models 300 to 800, 1-in. on Model 1200, 1¼-in. on Model 1600 for pressures up to 25 lb.

For pressures not exceeding 125 lb., a thermostatic trap of proper construction can be used and should be attached directly to the unit: same sizes as above except a ½-in. trap on Model 300 and 400, 1-in. trap on Model 1600.

CAPACITIES

60° F. Entering Air Temperature—5 lbs. Steam Pressure

Model Nos.	B.t.u. per Hour	Model Nos.	B.t.u. per Hour	Model Nos.	B.t.u. per Hour
100	24,500	100A	18,000	......	......
200	32,900	200A	24,000	......	......
300	82,300	300A	56,900	300B	72,300
400	110,800	400A	73,400	400B	94,000
600	159,500	600A	105,000	600B	136,500
800	199,100	800A	123,000	800B	166,500
1200	311,000	1200A	207,000	......	......
1600	388,000	1600A	246,000	......	......

Data Book covering other pressures and temperatures, dimensions and complete installation information on application. Address Grinnell Company, Inc., 277 West Exchange Street, Providence, R. I.

GRINNELL ADJUSTABLE PIPE HANGERS AND SUPPORTS

One of the chief advantages of Grinnell Adjustable Hangers is that they permit adjustment of pipe lines after installation, thus obviating the necessity of turnbuckles or the removal of hangers. Their time and trouble-saving qualities during installation are equally exceptional. Below are shown a few Grinnell Hangers and Supports of particular interest to heating engineers. Send for Hanger Catalogue showing complete line.

Fig. No. 101
Solid Ring

Fig. No. 104
Split Ring

Adjustable Swivel Rings (Patented)

These Malleable Iron Adjustable Swivel Rings can be used with Coach Screw Rod or Machine Threaded Rod in connection with practically any type of Ceiling Flange, Expansion Case, Insert, etc.

Adjustment of at least 1½ in. is secured by turning Swivel Shank. Swivel Shank automatically locks, preventing loosening due to vibration in the pipe line.

The off-center hinging of Split Ring provides sufficient seating to hold pipe line securely, and permits adjustment either before or after ring is closed. A wedge type pin is loosely but inseparably cast into the hinged section for fastening this section after pipe is in place.

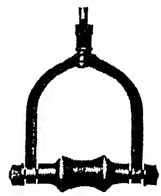


Fig. No. 174

Adjustable Swivel Pipe Rolls (Patented)

These Rolls supply the need for an adjustable type of pipe roll hanger using a single hanger rod. Vertical adjustment is made by use of Swivel Shank which automatically locks, preventing loosening due to vibration in the pipe line.

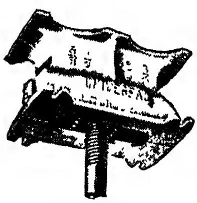


Fig. No. 280

Universal Concrete Inserts (Patented)

Made of malleable iron, they have ample vertical and horizontal adjustment. Made in one body size to take a special removable nut. Nuts furnished tapped for ⅜ in., ½ in., ⅝ in., or ¾ in. rod as required. Nuts automatically lock laterally by means of serrated teeth on both insert and nut.

U. F. S. I-Beam Clamps (Patented)

The U. S. F. I-Beam Clamps are an out-standing development in convenience, strength and adjustability. While made only in three sizes, they cover the whole range of beam sizes from the smallest to Bethlehem 24-in. Forged steel construction gives to each size the same strength as the maximum rod strength involved. Rods carried range from ⅜ to 1½ in., providing for all pipe sizes up to 24 in.

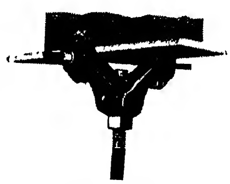


Fig. No. 228

Adjustable Pipe Roll Stands, Anchor Chairs, Brackets

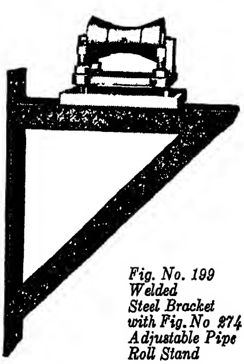


Fig. No. 199
Welded
Steel Bracket
with Fig. No 274
Adjustable Pipe
Roll Stand

The Grinnell Welded Steel Brackets, Fig. No. 199 are designed for use with Grinnell Adjustable Pipe Roll Stands, Fig. No. 274 and Anchor Chairs, Fig. No. 197, here illustrated. These combine the strongest type of brackets and pipe supports.

Vertical adjustment is obtained by adjusting screws on Stand; lateral movement by Stands sliding on ends of adjusting screws.

With Anchor Chair, lateral adjustment is possible if moved before nuts on yoke are tightened.

Fig. No. 197. Anchor Chair

GRINNELL WELDING FITTINGS

A Recent Major Addition to the Grinnell Line of Piping Supplies

The principal types of Grinnell Welding Fittings are illustrated and briefly described herewith. Catalogue giving list prices, dimensions, etc., will be sent upon your request.

Welding Elbows and Returns

90° Elbows, Long Turn
Standard weight
Extra heavy weight

45° Elbows, Long Turn
Standard weight
Extra heavy weight

180° Returns

Short Turn
Standard weight
Extra heavy weight

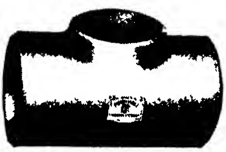
Long Turn
Standard weight
Extra heavy weight

Grinnell Welding Elbows and Returns, have the following distinguishing features:

1. Made from seamless steel tubing.
2. Uniform thickness of inner and outer walls.
3. Wall thickness same as steel pipe.
4. Homogeneous metal structure throughout.
5. No buckling or deformation.
6. No flattening of cross section.
7. Conformity to standard and extra strong pipe sizes.

Fittings can be cut on the job, straight across the pipe, to any desired angle quickly and accurately, and any simple or compound bend may be made by a combination of two or more fittings.

The process of manufacturing Grinnell Welding Elbows and Returns is not a bending process and is protected by U. S. and Foreign Patents.



Welding Tees

Welding Tees

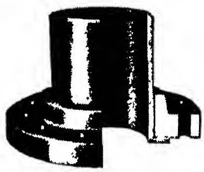
(Patents Applied For)

Grinnell Welding Tees, being of the same physical characteristics as Standard, Extra Strong and O. D. Steel Pipe or Seamless Steel Pipe of comparable pipe size, can be used under same conditions, pressures, and temperatures as the pipe itself. They have these distinctive features: Made from Seamless pipe; one piece fitting—no seams; outlet presents a smooth rounded surface.

Lap Flanged Welding Necks

Grinnell Lap Flanged Welding Necks are regularly furnished for working steam pressures of 150, 300, 400 and 600 pounds at total temperatures up to 750° F., and can be furnished on order for higher pressures.

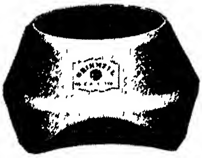
Each Welding Neck consists of a forged steel flange and a short piece of steel pipe or tubing with square lap having a thickness of at least 100 per cent of pipe walls. Opposite end of neck is scarfed for welding.



Lap Flanged Welding Necks

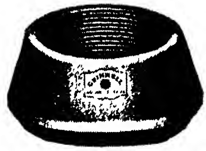
Welding Outlets and Threaded Outlets

(Patents Applied For)



Welding Outlets

Grinnell Welding Outlets and Threaded Outlets are stock fittings properly designed for pipe welding and provide a simple and economical method of making welded branch connections. They can be used wherever it is desired to make tees, side outlet tees, crosses, etc., by welding.



Threaded Outlets

Branches
AKRON, OHIO
BIRMINGHAM, ALA.
BOSTON, MASS.
CHICAGO, ILL.
DENVER, COLO.
DETROIT, MICH.

The Bristol Company

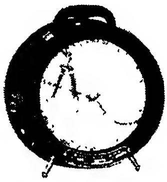
Waterbury, Conn.

Indicating, Recording and Control Instruments Since 1889

Branches
LOS ANGELES, CALIF.
NEW YORK, N. Y.
PHILADELPHIA, PA.
PITTSBURGH, PA.
ST. LOUIS, MO.
SAN FRANCISCO, CALIF.

TRADE MARK
BRISTOL'S
REG.U.S.PAT. OFFICE

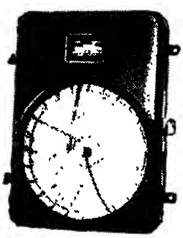
Direct Reading Relative Humidity Recorders



Model 4069

Bristol's Humidigraph, Model 4044, reads percentage of relative humidity directly. It gives a permanent chart record for future reference, and shows at a glance the trend of humidity condition. Simple construction. Novel vapor-sensitive hygroscopic element. Special aging process assures sustained accuracy. Eliminates calculation and use of humidity tables. Eliminates errors from personal element. No water required. No fan used. Accurate below freezing temperature. Light portable case of corrosion-resisting materials. Model 4069 is a combination Portable Humidigraph and Temperature Recorder, furnished with two pens for recording both relative humidity and temperature on same chart.

Recording Thermometers

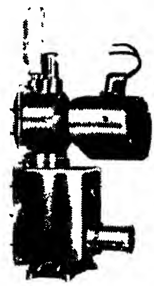


Model 240M

Class I Thermometers are either liquid-filled with connecting tube and bulb, or bimetallic self-contained. They are for ranges from −40°F to 150°F. Class II Thermometers are of the vapor tension type, and are used for ranges from 90°F to 650°F. Class III are gas-filled, for ranges from −60°F to 1000°F.

All three classes are furnished in a handsome moisture-proof, fume-proof, dust-proof rectangular case. One or more pen arms, upright or inverted. 12 in. or 8 in. chart, obtainable in over 800 ranges and graduated for one revolution in 24 hours or 7 days. Electric motor or spring wound clock. For wall or switchboard mounting, or portable.

Metameter Recording Telemetering System

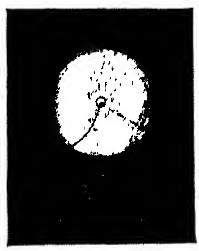


Bristol's Metameter Transmitter, side view

Bristol's Metameter System transmits instantaneously changes in pressure, temperature or liquid level occurring at a given locality to another locality as far as 240 miles away, and at this second locality produces an accurate undistorted graphic chart record of the quantity measured. The System consists of a transmitter, a recording receiver, a relay and rectifier box at the receiver, and a two wire circuit connecting transmitter and receiver.

Bristol's Metameter offers many outstanding advantages. Distance is limited only by the cost of installing or securing the necessary wires. Simplicity of circuit makes operation possible on only two wires, or on one wire and the ground. Over long distances, the impulse may be sent through arrangement with the A. T. & T. Co., over a standard telephone circuit also carrying conversation. Only 40 milliamperes at 6 volts D.C. are required for operation. Transmitter and receiver may use separate sources of power, as long as accurate timing is provided at each end. Variations in capacitance, in resistance, or induced currents do not introduce errors. No fire or explosion hazard. Reliable and durable mechanism.

Long Distance Electrical Transmitting and Recording System for Telemetering Steam Pressure



Long Distance Receiving Recorder, Model 340 MFR

Bristol's Long Distance Electric Transmitting System operates on the induction balance principle. Widely used by district steam heating companies, public utilities, etc., it provides centralized control of steam pressure at distant points throughout the distribution system.

Consolidated Ashcroft Hancock Co., Inc.

Bridgeport, Conn.

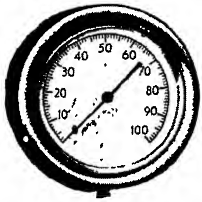
BRANCHES IN PRINCIPAL CITIES

Makers of AMERICAN INDUSTRIAL INSTRUMENTS—Since 1851

Subsidiary of Manning, Maxwell and Moore, Inc.

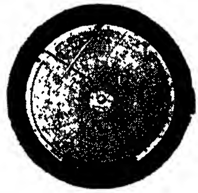
Manufacturers of Indicating and Recording Gauges; Gauge Testers; "U" Gauges; Draft Gauges; Indicating and Recording Thermometers; Tachometers; Dial Thermometers; Pressure and Temperature Controllers; Electric Temperature Controllers; Pop Safety and Water Relief Valves; Steam Traps; Engine Indicators; Counters; Absolute Pressure Gauges.
Also manufacturers of Bronze, Cast Steel and Forged Steel Valves, Locomotive and Engine Room Clocks; Barometers; Mercury Column Gauges; Steam Whistles; Hydraulagraphs; Gauge Boards.

Ashcroft American Gauges—Ashcroft American Gauges are made in all sizes from 2½ to 12 in., for pressures from 8 oz. to 25,000 lbs. and also for vacuum. Cases are cast-iron or cast brass. The movements are Heavy Duty and all bearings are Monel Metal. Write for Catalog No. A-59.



For Mercury Pressure and Vacuum Gauges, "U" Gauges, Draft Gauges and Mercurial Barometers, write for Catalog B-59.

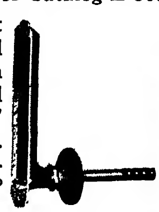
American Recording Gauges—American Recording Gauges are made for all pressures from 15 in. of water to 10,000 lbs. and for vacuum. They are made in one size only to accommodate a 10 in. chart, having an effective scale width of 3⅝ in. The case is Die Cast with a dull black hard-rubber finish and with either bottom or back connection. The pen-arm is made of non-corrosive Monel Metal and is of the inverted type. Operating instructions are lithographed on the chart plate so that they cannot be lost.



Especially designed Seth Thomas clocks are used, and all customary time periods can be furnished.

American Recording Gauges are equipped with the Time Punch which virtually makes each instrument a time clock, since a hole is punched in the chart whenever a reading is taken. Write for Catalog E-59.

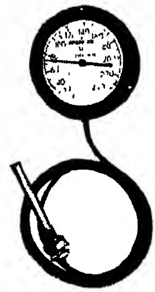
American Air Duct Thermometer—Designed especially for both warm and cold air ducts. Fitted with polished brass "V" shaped case, glass front. Furnished with 9-in. or 12-in. scale graduated 0-160° F. Write for Catalog F-59.



American Recording Thermometers—Made for recording temperatures from minus 40 to plus 1000° F. or equivalent C. Very flexible connecting tubing up to 200 ft. One size only to accommodate 10 in. chart, with an effective scale width of 3⅝ in.

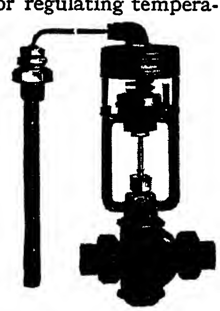
Same case as for the American Recording Gauge, so that all instruments are uniform in appearance when mounted on Gauge Boards. American Indicating Gauges and Dial Thermometers are also furnished in same case. Write for Catalog H-59.

American Dial Thermometers—American Dial (mercury-filled) Indicating Thermometer has the accuracy of the standard glass tube thermometer and the reading convenience of a dial face. Entire working mechanism is made of steel, meaning long life.



Standard size of dial 6 and 12 in. Furnished with rigid connection or flexible capillary steel tubing up to 200 ft. long. For temperature ranges from minus 40 to plus 1000° F. Write for Catalog G-59.

American Precision Temperature Controllers—Self-operated and simple in construction. For regulating temperatures from 25° to 385° F. Under favorable conditions temperature will be held within 1°. Sensitive, rugged and accurate. For hot water service tanks, water heaters, etc. Standard ranges carried in stock. Size of valve must be specified. Write for Catalog R-59.



Taylor Instrument Companies

Rochester, N. Y., U. S. A.

IN CANADA—TAYLOR INSTRUMENT COMPANIES OF CANADA, LTD., TORONTO

NEW YORK	PHILADELPHIA	LOS ANGELES	ST. LOUIS	DETROIT
CHICAGO	PITTSBURGH	INDIANAPOLIS	CINCINNATI	ATLANTA
BOSTON	CLEVELAND	SAN FRANCISCO	TULSA	MINNEAPOLIS

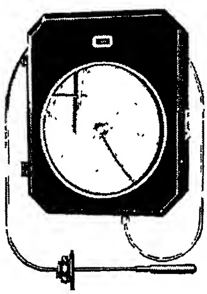
Manufacturing Distributors in Great Britain, Short & Mason, Ltd., London

Manufacturers of Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure and Humidity

Taylor

Taylor Recording Thermometers—Temperature ranges and time requirements vary greatly in heating and ventilating work. Taylor Recorders are made in scale ranges and time periods to meet these needs.

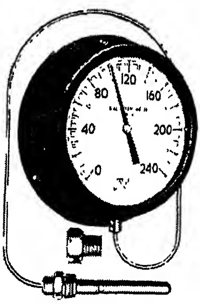
In their handsome new cases, these instruments are beautiful and efficient, particularly adapted for heating and air conditioning applications. They may be had for surface or flush mounting. When set in panel boards, the polished flanges make an effective installation.



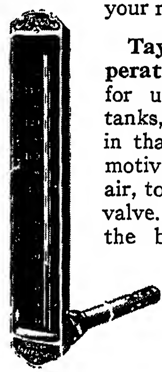
Write for special information suitable to your needs.

Taylor Electric Contact Temperature Control—These instruments combine in the same case an electrically-operated temperature regulator with an indicating thermometer. One tube system operates both units.

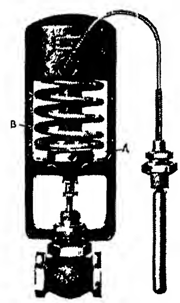
Taylor Dial Thermometers can be used for air ducts or any application where it is desirable to have temperature readings at some distance from the thermometer bulb, as in a central control room. Can be read at a glance as easily and quickly as a clock or steam gage.



Taylor Thermometers for Air Ducts, Etc.—The Taylor line of industrial thermometers presents many styles and scale ranges with bulbs for every application. Suitable for air ducts, kiln temperatures and oven temperatures. For detailed information, write direct, mentioning your requirements.



Taylor Self-Acting Temperature Regulator—Adapted for use on hot-water storage tanks, etc. It is "self-acting" in that it requires no auxiliary motive power, such as compressed air, to open and close the steam valve. As heat is applied to the bulb, the volatile liquid inside sets up a vapor pressure proportional to the temperature. This pressure is transmitted to a "stack" of metal diaphragms attached to the upper end of the valve stem, thus moving the valve disc toward the valve seat.

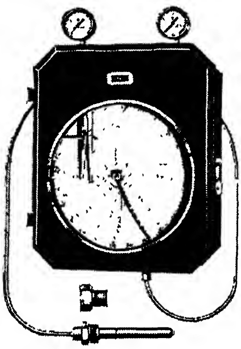


The valve can be closed at any desired temperature, or a throttling action can be obtained. Not practicable on pipe lines having steam pressure over 125 lbs. Should be installed in a vertical position on top of a horizontal line.

Operating ranges 120 deg. to 170 deg. F., 130 deg., to 190 deg. F., or 170 deg. to 240 deg. F. as specified.

Taylor Sling Psychrometer—The advantage of this form of Wet-and-Dry-Bulb Hygrometer over the stationary form is the facility with which tests can be made and the accuracy of the readings obtainable, as in whirling the bulbs they are subjected to perfect circulation. Consists of two accurate etched stem thermometers mounted on a die-cast frame.

The New Taylor "Fulscope" Recording Regulator—An air-operated recording regulator so versatile that *practically any character of process control can be obtained, regardless of *time lag* in apparatus, by a simple screw driver adjustment on a graduated dial—without requiring a skilled operator or interruption of service.



Vast improvements in entire mechanism.

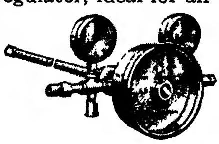
Easily changed from direct to reverse-acting, or *vice versa*—no extra parts.

Compensates for fluctuations in air-pressure supply.

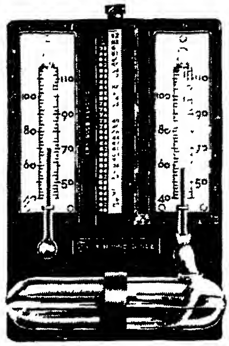
Die-cast case; dust-, moisture-, and fume-proof.

Available forms: for controlling temperature, pressure, temperature and pressure, rate of flow, liquid level.

Taylor Type-P Regulator—A compact and very sensitive regulator, ideal for air-ducts, air-washing machines, cooling rooms and similar applications. Uses compressed air as an actuating medium.

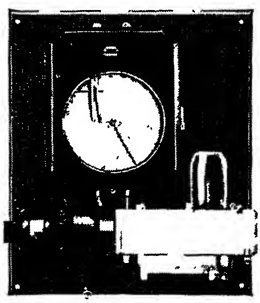


*Where extreme load changes or badly balanced operating conditions exist, the *Taylor "Dubl-Response Control Unit"* is the only positive means of maintaining control-point. *Write for literature.*



Taylor Humidiguide—A handsome small hygrometer for the wall of the home, office, school or other building where a neat, easy-reading and inexpensive instrument is desired. It is self-contained, requiring no charts or separate tables. Frame is Mahogany Bakelite.

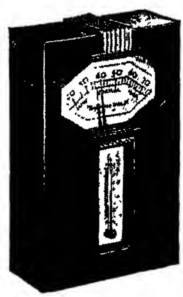
Taylor Recording Hygrometer—This instrument records both wet- and dry-bulb temperatures on the same chart in different colored inks, making comparison very easy.



Type shown above with motor-driven fan for conditioned rooms or passages in which circulation is poor. Can be supplied without fan for installations where circulation across bulb is good.

Taylor Anemometer—This instrument is ideal for measuring air velocities with the fan revolutions indicated on the dial. Available in various models for a wide range of air speeds and registration limits.

Taylor Hampton-Model Humidiguide (Direct - Reading)—A hygrometer giving direct humidity percentages, in a smart modern case suitable for home, office or public building. Finish is satin black with chrome trim. The thermometer has a Permacolor tube, distinct, non-fading red column.



Alfol Insulation Co.

Chrysler Building :-: **New York, N. Y.**

Agents in Principal Cities

INSULATION *for*

Fans · **Blowers** · **Pumps** · **Ducts** · **Houses, Buildings, etc.**
Turbines · **Dehumidifiers** · **Air Conditioners** · **Boilers, Pipes, etc.**

At temperatures up to 1150° F.

ALFOL

Patented in 34 Countries

INSULATION *for*

Refrigerators · **Refrigerator Cars** · **Refrigerator Trucks** · **Refrigerator Boxes** · **Refrigerated Rooms**
Ships · **Ovens** · **Ranges** · **Tanks** · **Stills**
etc.

DESCRIPTION

Alfol Insulation-layers spaced approximately 3/8″ apart
Conductivity—Panel Type .22 Crumpled Type .28

ALFOL Insulation is a flexible form of insulating material for use on irrigular, curved or straight surfaces where it is desirable to conserve heat or cold, prevent condensation and block sound transmission. In its simplest form it consists of successive layers of crumpled special metal foil applied approximately 3/8″ apart. The outside layer of ALFOL is covered with a removable metal jacket.

ALFOL is a modern, scientifically designed insulation which effectively and economically meets all high and low temperature requirements. Some of the many advantages are:

Alfol Metal-Jacketed Pipe Lines

1. **Negligible Weight**—only 1/4 oz. per board foot.
2. **Highly Efficient**—conductivity .28 B.t.u. per hour, per sq. ft., per 1° F temp. difference, per inch thick.
3. **Non-Inflammable**—being metal it will not burn.
4. **Impervious to Moisture**—repels all moisture attacks thereby retaining its uniform high insulating value.
5. **Durable** — unaffected by vibration, will not disintegrate, crumble, warp or settle.
6. **Odorless and Clean**—ALFOL has no odor. No dust, dirt or waste result from its use.
7. **Low Heat Storage**—ALFOL possesses virtually no heat storage capacity—therefore pre-heating or pre-cooling may be done much more rapidly than with any other insulation.
8. **Low Initial Cost**—ease, rapidity of application and no dirt or waste make the initial cost of ALFOL lower than for most other insulations.

APPLICATION

ALFOL Insulation is applied in never more than 3 crumpled layers to one inch thickness. The high re-reflectivity of the foil (95%) and the non-conducting air spaces between the layers, provide ALFOL's amazing efficiency. It is never necessary to wad or pack ALFOL in place. As a matter of fact, when applied loosely in corners, slots, etc., its efficiency is actually increased. All ALFOL joints are lapped in each layer about 1″ and staggered with respect to joints in adjacent layers. All stiffeners, studs, rivet heads, etc., protruding into the insulation are lightly covered over with the same number of layers used on surrounding surfaces.

Alfol Insulation for Heating and Ventilating Ducts

Removable metal jackets with their many obvious advantages are applied over ALFOL Insulation.

Alfol Insulation on 9000 KW. Turbine

SIZES

ALFOL is furnished in convenient size rolls of 2000 or 3000 sq. ft. Rolls are approx. 6″ diam. x 15 3/4″ long.

ENGINEERING

Trained ALFOL engineers are ready to assist in the solution of your insulating problems.

Texas & Pacific R. R. Alfol Insulated Train

(See Chapter 5 for information on Aluminum Foil Insulation)

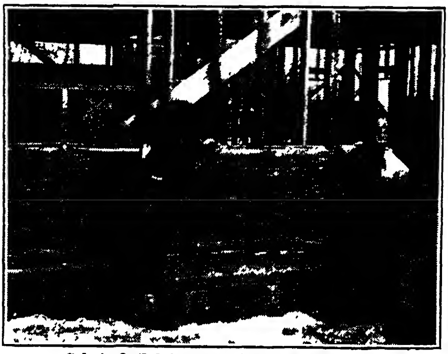
141 Milk Street **Samuel Cabot** *Incorporated* **Boston, Mass.**

101 Park Avenue, New York
5000 Bloomingdale Avenue, Chicago

Philadelphia, Kansas City, Minneapolis, Los Angeles, San Francisco, Seattle, Portland

Agents in All Principal Cities

Cabot's Heat-, Cold-, and Sound-Insulating Quilt



Cabot's Quilt being applied to under side of roof and walls, for heat insulation

Uses—For insulating side walls and roofs of buildings against cold in winter and heat in summer, for insulating cold storage buildings and ice houses, refrigerators, etc.; for sound-proofing partitions and ventilating ducts.

Description—Cabot's Quilt is a felted springy matting of Zostera Marina, a marine plant, stitched between two layers of Kraft (Standard Quilt), non-inflammable Kraft (Anti-Pyre Quilt), asbestos (Asbestos Quilt), water-proof paper (Water-proof Quilt). Cabot's Quilt has an insulation value inch for inch excelled or equalled by only two other insulators, both greater in cost.

Permanent—Because of its natural properties, no special chemical treatment is required, and Cabot's Quilt will not pack down, rot or harbor insects or vermin. Quilt shows no deterioration after having been in actual use for over forty years.

Fire Resistance—Standard Quilt is naturally fire resistant and when made with the new non-inflammable Kraft paper (Anti-Pyre Quilt) can be used in any fire-proof building.

Easy, Low Cost Application—Quilt can be applied at the lowest labor cost. Being flexible it fits into corners or around projections without danger of cracking. Requires no special preparation or smoothing of walls or floors in cold storage work. Easily adapts itself to lining or covering of ventilating ducts. Readily cut with ordinary knife or shears.

Widths—Quilt is made in 36″ and 18″ widths. 18″ Quilt is quickly applied between studs and roof rafters.

Thicknesses—Cabot's Quilt is made in Single-Ply (X), about 4/10″ thick. Double-Ply (XX), about 6/10″ thick. Triple-Ply (XXX), 8/10″ thick; and Inch Quilt, 1 2/10″ thick.

U. S. Government Bureau of Standards Letter Circular No. 227, reporting tests of numerous Insulating Materials, gives **Cabot's Quilt** a Thermal Conductivity rating of 0.25 per inch, which was equalled by only two other insulators, regardless of cost.

Tests conducted by Prof. Gordon B. Wilkes, of the Massachusetts Institute of Technology, show the following savings in heat leakage in different methods of construction, by the use of **Cabot's Quilt.**

CONSTRUCTION WALL	Conductivity Uninsulated "H"	Conductivity Insulated with Cabot's Double Ply Quilt—"H1"	Percentage Heat Saving
Clapboard, studding	0.70	0.26	63
Clapboard, studding, lath, plaster	0.44	0.21	52
Clapboard, sheathing, studding, lath, plaster	0.28	0.16	43
8 in. brick	0.40	0.20	50
8 in. brick, furring, lath, plaster	0.27	0.16	41
4 in. tile, stucco, outside, plaster inside	0.40	0.20	50
12 in. concrete	0.46	0.21	54
Corrugated iron	1.50	0.32	79
12 in. stone	0.49	0.22	55
12 in. concrete, furring, lath, plaster	0.40	0.20	50
Stucco, studding, plaster	0.45	0.21	53
ROOF			
Tar, gravel on 4 in. concrete	0.60	0.24	60
Metal on tongue and groove sheathing	0.42	0.20	52
Corrugated iron on wood frame	1.80	0.32	82
Tile or slate on wood sheathing	0.82	0.27	67
Tar, gravel, tar paper, 2 in. wood plank	0.26	0.16	39
Shingles, sheathing, studding, lath, plaster	0.30	0.17	43
Wood shingles on shingle lath	0.64*	0.24	63

*Average value from Jones' tables, *The Heating and Ventilating Magazine.*

The cost of Cabot's Quilt is so low, for material and application, that the resulting smaller Heating Equipment required usually more than pays for entire insulation. Thereafter there is a yearly fuel saving.

PRODUCTS—

- **Building Board.**
- **Lath, ½", ¾" and 1".**
- **Sheathing, ½", ¾" and 1".**
- **Tile Board.**
- **Industrial Insulation Board.**
- **Roof Insulation Board.**
- **C-X Wallboards.**
- **Low Temperature Insulation.**
- **Rock-Wool-Batt.**
- **Acousti-Celotex Cane Fibre Tile.**
- **Acousti-Celotex Mineral Fibre Tile.**

Celotex Cane Fibre Insulation

Celotex is a rigid insulation manufactured by felting strong, tough cane fibres into a continuous board. Several products are then fabricated for various purposes. The resulting products combine high insulating efficiency with unusual structural strength.

The thermal conductivity of Celotex is 0.33 Btu per hour, per square foot, per 1 degree Fahrenheit, per inch thickness (based on a density of 13.5 lb. per cubic foot and a mean temperature of 70° F.) Tests conducted at Armour Institute of Technology and at recognized laboratories confirm this figure.

The Celotex Company maintains an engineering and research staff which is available for investigations of all types of insulation installations. Engineers are invited to address their problems to The Celotex Service Bureau at Chicago.

The Ferox Process

The Ferox Process (patented) is exclusive with Celotex. It makes all Celotex cane fibre products safe from dry rot and termites—proof against those two age-old enemies of wood and other cellulose materials which cause unbelievable damage to buildings (about five million dollars a year in Illinois alone) in the United States each year. By means of this process—perfected after ten years of research and two years of actual production—Celotex provides a powerful protective armor against loss through deterioration.

The Ferox Process is not a surface treatment—it is integral. The chemical complex used is insoluble in water. It is non-volatile — odorless — permanent. It does not discolor the product or otherwise alter its physical properties. It has been tested out successfully for over two years in the tropics where the termites are so active that entire buildings are sometimes ruined in the course of a few weeks.

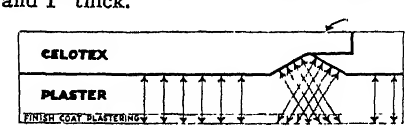
Celotex Building Board

Two-surface utility. One side smooth for stenciling and other interior finish treatments. The other retains that unique Celotex texture. Sizes—4 feet wide, and 7, 8, 8½, 9, 9½, 10 and 12 feet long. 7/16" and ⅞" thick—also ½" and 1" thick with both sides rough texture.

Finish Plank—7/16" thick; 6", 8", 10", 12" and 16" wide; 8' long. Long edges Beveled and Beaded.

Celotex Lath

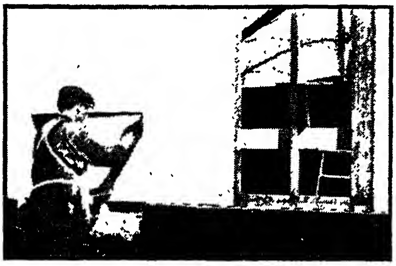
A natural bond for plaster—a continuous plastering surface providing special resistance to lath cracks—eliminated lath marks—beveled and shiplapped joints (see diagram). Size: 18" x 48". ½", ¾" and 1" thick.



Celotex Sheathing

Insulation and structural strength. May be used with standard frames. ½″, ¾″, 1″ thick; 4′ wide; 7′, 8′, 8½′, 9′, 9½′, 10′ and 12′ long.

Sheathing Plank—¾″ thick; 18″ wide; 8′ long.



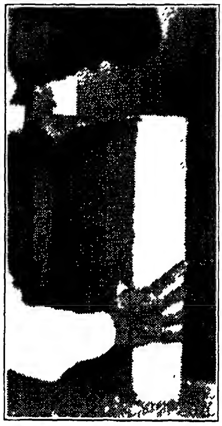
Celotex Tile Board

An attractive interior finish which insulates against heat and cold and deadens noise. For installation in new and old buildings alike. Has a smooth surface like Celotex Building Board, which may be left natural, or may be painted or stenciled. Thicknesses, 7⁄16″ and 13⁄16″, ranging in sizes from 6″ x 12″ to 24″ x 32″.

Celotex Low Temperature Insulation

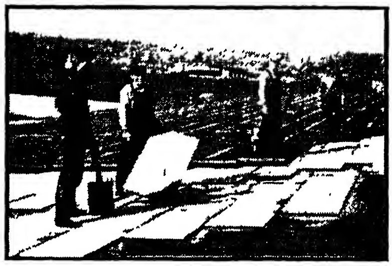
A moisture-proof, vapor-proof low density insulation. Each block hermetically sealed by means of a coating of asphalt and a wrapping of two thicknesses of heavy waterproof paper with asphalt between, all edges being thoroughly sealed with asphalt. For all low temperature requirements, including Coolers (Beer, Meat, Creamery, etc.), Fruit and Vegetable Storage Rooms, Air-Conditioned Spaces, General Cold Storage Rooms, and Freezers. Conductivity 0.30 Btu per inch—odorless. Ferox-treated. Size 18″ x 36″. Thickness 1″, 1½″, 2″, 3″, 4″ or any multiple of ½″.

Celotex Rock-Wool-Batt



Highly effective wall-thick insulating material made from Molten Rock. Absolutely incombustible, vermin-proof and permanent. Light in weight (about 2½ lbs. per unit). Fits snugly between studs and rafters. Size 15″ x 18″ x full wall-thickness.

Celotex Roof Insulation Board



Preferred as insulation over wood, concrete, steel, unit tile and poured gypsum roof decks. Sizes 22″ x 46″ approximately ½″ thick. Also furnished laminated from 2 to 8 plies.

Service

The Celotex Company has sales distributors throughout the world. This world-wide service has a background of trained insulation engineers who are at your service whenever some unusual insulating problem is before you. Write to the Celotex Service Bureau, 919 North Michigan Avenue, Chicago, Ill., for information of any kind, without obligation.

Johns-Manville

Executive Offices

22 East 40th Street, **New York, N. Y.**

Offices in All Large Cities

Johns-Manville Home Insulation

J-M Home Insulation involves the application of J-M Rock Wool (actual rock fibre) within the walls and the roof or attic spaces of residences, stores, apartment buildings and other structures.

J-M Home Insulation is *thick* insulation, remarkably effective in providing year round comfort and in reducing fuel bills. (On hot summer days homes insulated with this material are up to 15° cooler; in winter fuel bills are reduced from 25% to 40%). J-M Home Insulation is non-combustible, sanitary and odorless, and will not support vermin.

Furnished in two forms: Type A for blowing into existing construction; and Type B bats for new homes.

Blown Method—Type A

In the Type A form, Rock Wool is blown by air into the spaces between studs in outer walls and between rafters or joists in attic floors. The insulation thickness in the walls corresponds to stud depth, approximately 3⅝ in., and the density does not exceed 10 lbs. per cu. ft. J-M Home Insulation has been installed by this method in thousands of existing homes.

This type of material is installed by J-M Approved Home Insulation Contractors, who are equipped with the necessary apparatus.

Applying J-M Home Insulation bats in new home

Bat Method—Type B

Home Insulation Type B is furnished in resilient bats, 15 in. by 18 in., approximately 3 in. to 4 in. thick. This form is widely used for homes or buildings under construction. The bats can be readily pressed between studs or beams and are easily cut or torn to fit odd-shaped spaces, conforming perfectly to the space occupied and providing a continuous insulation of even density.

Data and Specifications

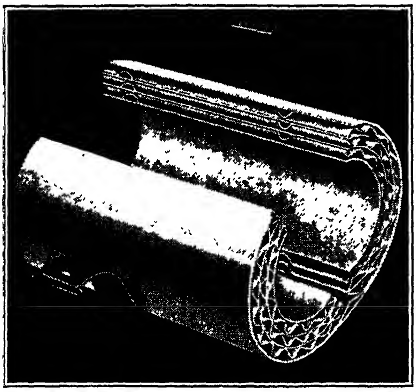
For technical data and specifications on both types of J-M Home Insulation, write for brochure HI-17A.

J-M Insulating Board and Plaster Lath

J-M Insulating Board is a light weight, efficient material, having high insulating value and moisture resistance and unusual structural strength and rigidity. Furnished 4 ft. wide, in lengths of 4, 5, 6, 7, 8, 9, 10 and 12 ft. Also 6 by 8, 8 by 8, 8 by 10 and 8 by 12 ft. Thicknesses ½ and 1 in.

J-M Insulating Lath is the same material as Insulating Board, except that it is furnished 18 in. by 48 in. with long edges ship-lapped. Also furnished with long edges ship-lapped and beveled, and short edges beveled. Thicknesses ½ in. and 1 in.

Johns-Manville Pipe and Boiler Insulation



J-M Pre-Shrunk Asbestocel Pipe Insulation with aluminum finish

J-M Pre-Shrunk Asbestocel

J-M Pre-Shrunk Asbestocel is a radically improved material in which, by the use of waterproofed asbestos paper, shrinkage troubles have been eliminated. It is used for hot water or low pressure steam piping, including supply and return mains, branches and risers.

Supplied in three finishes: The regular canvas finish; and the new high-speed asbestos paper or aluminum finished material which slips easily over the pipe and clinches on tight with quick-fastening staples.

All types are furnished in 3-ft. sections in standard thicknesses of 2, 3, and 4 plies, each ply approximately ¼ in. thick.

J-M 85% Magnesia

Recommended as the most efficient insulation of the molded type for temperatures up to 600 deg. F. Pipe insulation is furnished in sectional or segmental form for all commercial pipe sizes, in thicknesses up to 3 in. Blocks are 3 in. by 18 in. and 6 in. by 36 in., flat or curved, from ½ in. to 4 in. thick.

J-M Wool Felt

Due to its Dual-Service Liner—an asphalt-saturated felt—J-M Wool Felt is equally effective and durable on either hot or cold water service piping. By the use of waterproofed felts shrinkage troubles have been eliminated.

Supplied in two finishes, the regular canvas and a smooth, dull-coated aluminum. In either finish, it is furnished in 3-ft. sections in thicknesses of ½ in. ¾ in., 1 in., Double ½ in., and Double ¾ in., for pipe sizes from ½ in. to 5 in.

J-M Asbesto-Sponge Felted

Recommended on all high pressure steam piping at temperatures up to 700 deg. F. where insulation may be subjected to rough usage or where maximum efficiency and durability are desired. Furnished in 3-ft. sections up to 3 in. thick.

J-M Superex Combination

Superex Combination Insulation (an inner layer of high temperature Superex and an outer layer of 85% Magnesia) is recommended where temperatures exceed 600 deg. F. Superex and Magnesia are both furnished in sectional and block forms.

J-M Asbestocel Sheets

Asbestocel Sheets are used for insulating warm-air ducts, flues, heater casings and fan housings in the ventilating system. Temperature limit 300 deg. F. Furnished 6 in. by 36 in. and 36 in. by 36 in., from ½ in. to 4 in. thick.

J-M Rock Cork

J-M Rock Cork is made of rock wool and a moisture-proof binding ingredient molded into sheets for insulating refrigerated rooms and air conditioning ducts. It is strong, durable, and will not support vermin. Because of its unusual moisture resistance its high insulating efficiency is maintained indefinitely.

Furnished 18 in. by 18 in. and 18 in. by 36 in., in thicknesses from 1 in. to 4 in.

Detailed Specifications

Specifications on the use of any J-M Insulating Material may be had on request.

International Fibre Board Limited

TEN/TEST — **Insulating Building Board**

PYRO/TEST — NON-INFLAMMABLE INSULATING BUILDING BOARD — **Non-Inflammable Insulating Building Board**

Sales Offices
OTTAWA—MONTREAL—TORONTO—WINNIPEG
Administrative Offices and Mills: GATINEAU, QUE.
London Office
THE TENTEST FIBRE BOARD CO. (1929) LTD.
ASTOR HOUSE, ALDWYCH, LONDON, W. C. 2., ENGLAND

TEN/TEST is a manufactured lumber made from spruce fibres, solidly pressed under hydraulic pressure into a strong, homogeneous board. The fibres are chemically treated and water-proofed during process of manufacture, until the insulation is non-hygroscopic, free from capillary attraction and moisture-resisting in service commensurate with the maximum degree of insulation obtainable.

Official Tests

Conductivity. TEN/TEST has a conductivity of 0.33 B.T.U. per hour per square foot per degree fahr. per 1 in. thick. Authority: Professor E. A. Allcut, M. Sc. M. I. Mech. E. Mem. A.S.M.E. Professor of Applied Mechanics, University of Toronto. Tests performed by Hot-Plate method. Mean temperature 47.8 deg.

Tensile Strength 228 lb. per sq. in. Tests made on $\frac{7}{16}$ in. board cut to strips 1 in. wide and tested in a Riehle Tensile Testing Machine, the grips being 2 in. apart. 228 lb. is the mean average of seven series of tests.

Transverse Strength (equal deflection) is 28.4 lb. Test made on $\frac{7}{16}$ in. board, 6 in. wide, 18 in. long, on 12 in. centers, and load being applied to breaking point.

Plaster Bonding Strength 2163 lb. per sq. ft. Brown and scratch plaster coats were applied to standard $\frac{7}{16}$ in. board, and the pull registered in an Olsen Testing Machine. Authority: Columbia University Testing Laboratories, New York.

Moisture Resisting. TEN/TEST, after complete immersion in water for 24 hours, registered 37.5% increase in weight.

NOTE.—Authority for tensile strength, transverse and moisture tests; J. T. Donald & Co., Ltd., Chemical Analysts and Engineers, Montreal, Que.

TEN/TEST Products

TEN/TEST Insulating Building Board. Standard insulation for use as exterior sheathing, interior finish; between walls and under floors for sound deadening. Standard Industrial Insulation for refrigeration and the prevention of condensation. Manufactured in convenient sizes: 4 ft. wide and up to 17 ft. long, $\frac{1}{2}$ in. to 2 in. thick.

TEN/TEST Notch Board Plaster Base. Insulating plaster base having tongue and groove interlocking joints. Provides an effective bond with plaster without use of metal lath at joints. Sizes: 16 in. wide; 32 in. and $47\frac{3}{4}$ in. long. Thicknesses from $\frac{1}{2}$ in. to 2 in.

TEN/TEST Roof Board. An effective roof insulation. Manufactured in two sizes: 1 x 4 ft. and 2 x 4 ft. Thicknesses from $\frac{1}{2}$ in. to 2 in.

TEN/TEST Ashlar Block (Acousti "A"). For interior decoration and acoustical correction. Absorbs 35 per cent of incident sound at a frequency of 512. Can be supplied in a variety of designs and sizes to harmonize with any decorative treatment, allowing the architect much freedom in design and finish of churches, auditoriums, theatres, etc. Ashlar Blocks have bevelled edges, standard or to suit, can be left in the natural color or tinted as desired.

TEN/TEST Panel Strip. Provides pleasing trim and finish for joints, corners and openings. Manufactured in widths of from 2 in. to 6 in. and lengths up to 12 ft.

PYRO/TEST. Non-inflammable, fire-resistant, insulating building board with same physical characteristics as TEN/-TEST.

HYDRO/TEST. Water proof, insulating building board, designed particularly for low temperature requirements.

Mundet Cork Corp.

450 Seventh Avenue — New York, N. Y.

Manufacturers of Corkboard, Cork Pipe Covering, Compressed Machinery Isolation Cork, Natural Cork Isolation Mats, Cork Tile, Cork Bulletin Board, and all kinds and varieties of Cork Specialties.

Branches

Atlanta, Ga.	Cincinnati, Ohio	Detroit, Mich.	Los Angeles, Calif.	Philadelphia, Pa.
Boston, Mass.	Cleveland, Ohio	Houston, Texas	Memphis, Tenn.	St. Louis, Mo.
Buffalo, N. Y.	Des Moines, Iowa	Kansas City, Mo.	New Orleans, La.	San Francisco, Calif.
Chicago, Ill.				Tulsa, Okla.

Agents

Charlotte, N. C.	C. R. Howard	Portland, Ore.	Pacific Asbestos & Supply Co.
Hartford, Conn.	The Hartford Cement Co.	Salt Lake City, Utah	L. A. Roser
Pittsburgh, Pa.	Smith-Doeright Co.	Seattle, Wash.	Pioneer Sand & Gravel Co.
Portland, Ore.	F. J. Leonard	Utica, N. Y.	George Weisenberger

Engineering and Specification Service

Our engineering department is at the service of Architects and Engineers at all times to assist and advise in the preparation of specifications pertaining to cork. This service is also available to any one who has a cold insulation or a vibration isolation problem, and is rendered without obligation. Send for our catalogue now. It is replete with valuable information and should always be within reach of every specification writer whose field touches our products.

Contract Service

We contract for the erection of our products in order that we may be certain that our material is installed in accordance with best established practice, and in order to eliminate divided responsibility for a given installation. No contract involving cork is too large or too small. All materials and workmanship are unqualifiedly guaranteed.

Mundet "Jointite" Corkboard

Mundet "Jointite" Corkboard is 100 per cent pure cork, fabricated in accordance with the U. S. Government Master Specification, and is unsurpassed in its field. It is used for all cold insulation services and for acoustical correction. We manufacture only one grade of corkboard. Mundet "Jointite" Corkboard is sold in the standard 12 in. x 36 in. sheet. Standard thicknesses are ½ in., 1 in., 1½ in., 2 in., 3 in., 4 in. and 6 in.

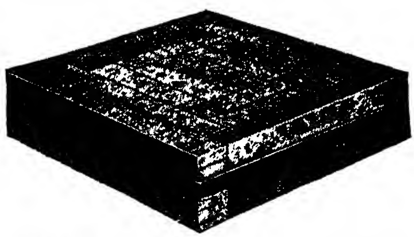
Mundet "Jointite" Cork Pipe Covering

Mundet "Jointite" Cork Pipe Covering is the complement of Mundet "Jointite" Corkboard and is used for all types of cold lines. The three thicknesses in which it is manufactured make it suitable for pipes carrying sub-zero to 50° temperature. The pipe covering comes in sections 36 in., long. A complete line of standard fitting covers is available in the three thicknesses.

Mundet Cork Vibration Isolation

Machinery isolation is today a most important part of the design of all structures. The transmission of machine vibration can be easily and permanently prevented by the use of proper cork isolation. Mundet Natural Cork Isolation Mats are best adapted to most of the isolation field and Mundet Machinery Isolation Cork to the balance.

Mundet Natural Cork Isolation Mats are blocks of pure cork. These blocks are held together within a rigid steel frame, or bound with asphalt paper applied with hot asphalt top and bottom. Mats are constructed to fit under any type of machine foundation.



Above is shown a typical Mundet Natural Cork Isolation Mat. Note the natural cork strips within the steel frame.

Mundet Machinery Isolation Cork is manufactured board and comes in the three standard densities. It is composed of granules of pure cork, fabricated and baked under pressure. The standard size board is 12 in x 36 in. but these can readily be cut into any size or shape.

Both types of isolation are furnished in 1 in., 1½ in., 2 in., 3 in., 4 in. and 6 in., thicknesses, depending on the class of service.

The Upson Company

Lockport, New York

UPSON *Insulating* BOARD

AND

UPSON *Insulating* LATH

Upson Insulating Board

Upson Insulating Board is a structural, rigid board form of Insulation, made from wheat straw, for use in walls, floors, ceilings and partitions. It quiets sound, saves fuel and adds comfort in all seasons.

Upson Insulating Board may be used as a sheathing under wood shingles, wood siding, masonry or stucco veneer.

Tests

Upson Insulating Board, in its commercial thickness, has a Resistance or Insulating Value of 1.36.

Conductivity per 1 in. of Thickness—.324.

Conductance of Commercial Thickness—1.74 B.t.u. per square foot per hour per degree Fahrenheit Temperature Difference.

Tension 1 in. wide Commercial Thickness—163½ lbs.

Tension, pounds per square inch—362 lbs.

Mullen Test Commercial Thickness—433½ lbs.

Modulus of Rupture—923 lbs.

Application

Upson Insulating Board is furnished 48 in. wide for application directly to studs, joists or rafters spaced 12, 16 and 24 in. on centers. Panels to be applied parallel to studs and joists.

Sizes

Upson Insulating Board is supplied in panels $\frac{7}{16}$ in. in thickness—48 in. wide and lengths of 6, 7, 8, 9, 10 and 12 ft.

Average weight is approximately 70 lbs. per 100 sq. ft.

Upson Insulating Lath

Upson Insulating Lath is a structural insulating plaster base. It is scientifically designed to meet the exacting needs and requirements for obtaining better plastered walls and ceilings.

The exclusive nail slot, and enclosed feathered edge spaces at all edges provide room for the expansion of the lath. These permanent expansion spaces protect the plaster from strains and stresses set up in the framing members, due to shrinkage.

The ship-lapped edges exclude infiltration of air at the joints, thus preventing unsightly streaks of dust deposits at these points.

Application

Upson Insulating Lath is furnished in panels 18x48 in. and is applied horizontally to studs and joists so that all vertical joints are staggered or broken. Only gypsum or other quick drying plaster should be used.

Sizes

Upson Insulating Lath is furnished in $\frac{7}{16}$ in. thickness, in individual lath 18x48 in. They are packed 15 lath to the bundle, each bundle containing 90 sq. ft. and weighing approximately 59 lbs. per bundle.

REG. U. S. PAT. OFF.

RIC-WIL

CONDUIT SYSTEMS FOR UNDERGROUND STEAM PIPES

The RIC-WIL Company

Union Trust Bldg. **Cleveland, Ohio**

NEW YORK—CHICAGO—SAN FRANCISCO

Agents in Principal Cities

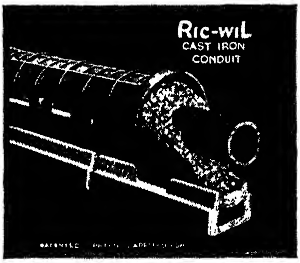
ESTABLISHED 1910

Conduit—Standard conduit is vitrified salt glazed tile or cast-iron with Loc-liP Side Joints, bell and spigot type, unlined or lined. Tile in 24 in. sections and cast-iron in 48 in. sections, sizes 4 to 27 in. inside diameter. Fully described in Bulletin No. 32. A special light weight cast-iron conduit with flanges, bolts and gaskets for bad water conditions and a heavy weight cast-iron conduit for extra heavy duty are also available. Send for A & E Sheets Nos. 14 and 23 for details of these special cast-iron conduits.

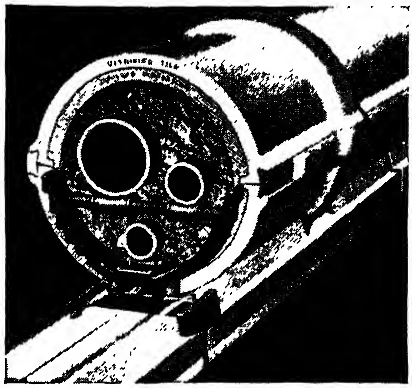
Base Drain—Standard Base Drain is vitrified salt glazed tile for tile conduit and extra heavy tile or cast-iron for the cast-iron conduit, in 24 in. lengths. Special U-Type Base Drain to hold return line and Support Blocks for use with round tile construction also available.

Pipe Supports—Standard Pipe Supports are of the externally supported type with rollers for single or multiple pipes. Load is carried entirely on side shoulders of Base Drain. Pedestal Pipe Supports are shaped like the Base Drain and set between Base Drain sections embedded in a concrete pier. Internal Pipe Supports also available. All Pipe Supports are rust-proofed. Details of Ric-wiL Pipe Alignment Guides and Anchors furnished upon request.

Insulation—Dry-paC Waterproofed Insulation, non-settling and non-corrosive —packed into the conduit becomes a solid mass without cracks, joints or openings—positively non-capillary and water-repellant. Has low conductivity and insures highest thermal efficiency—non-waterproof kind also furnished. Lined conduit insulation is a moulded diatomaceous earth mixture. Sectional pipe covering, sponge felt, 85% magnesia, etc., can also be furnished.



Ric-wiL Cast Iron conduit has ample strength with minimum weight for use under roadways, railroad tracks, or other places subject to extreme loads and vibration. It is installed without delays and without added construction or engineering.



Cutaway view at pipe support showing how Ric-wiL parts interlock. Note Loc-liP Side Joints and how pipe assembly is independent of the conduit. Cut shows Type DF System, lined conduit, with Dry-paC Waterproofed Insulation.

Accessories—Conduit fittings, with Loc-liP Side Joints for all types up to 15 in. are carried in stock—45° and 90° elbows, tee branches and reducers. Tee branches and reducers made to order for larger sizes. Mitred Base Drain fittings made for all conduit fittings. Shutter sleeves of galvanized iron, with asbestos rope for encircling pipes to provide a form upon which to cement or brick in ends of conduit lines. Filter Cloth to prevent sand, etc., from sifting thru broken stone into drainage area of Base Drain. Special asphalt cements for sealing all joints and waterproofing compounds for cement joints. Manhole covers, with non-rattling double lidded covers, furnished in 22 in. and 30 in. sizes.

Engineering Service—Cooperation in preparing preliminary surveys, plans and layouts or complete plans with full construction details. Also installation supervision.

Technical Data—Tabulated Steam Heating Rates, Test Report Bulletins, Service Detail Bulletins, Catalog Bulletins, Central Heating Bulletins and Architects and Engineers Detail Sheets available upon request.

Underground Steam Construction Co.

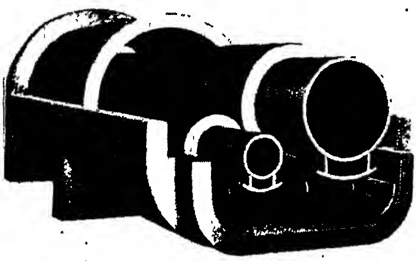
75 Pitts Street, **Boston, Mass.**

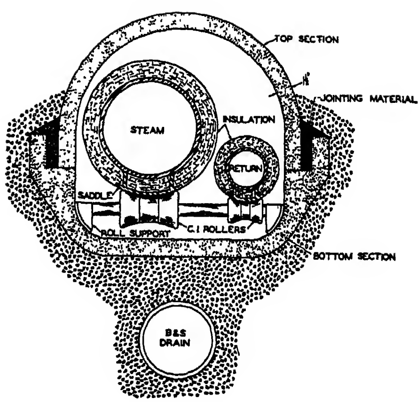
PRODUCTS—Underground Steam Conduits. Engineering and Contracting of Steam Line Installations.

USCCO PRE-CAST CONCRETE CONDUIT

This unit marks a long step forward in the economics and mechanics of laying underground steam lines.

Its Outstanding Features Are:





Strength resulting from flat reinforced bell; from the longitudinal joints, and from the fact its sections are 4 feet long, reducing the usual number of joints necessary.

Shape which allows 12 per cent greater inside capacity than circular conduit of like diameter. This permits larger pipes in a given size conduit and more room for drainage, pipe supports and rolls.

Ease of Installation resulting from the fact that **all** top and bottom halves mate. Bottoms may be laid for any distance, the piping installed and then the tops brought up and laid. This speeds up the work and protects idle halves from damage. They can be stored away from the job.

Materials are high strength cement with suitable aggregate, amply reinforced with wire mesh.

Pipe Supports are of cast-iron and quickly installed. They may be placed anywhere in the conduit. The weight of the pipe holds them in place.

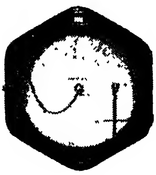
Joints may be of any standard accepted brand of jointing compound or they may be of standard Portland cement mortar.

Builders Iron Foundry

9 Codding Street **Providence, R. I.**

Representatives in Principal Cities

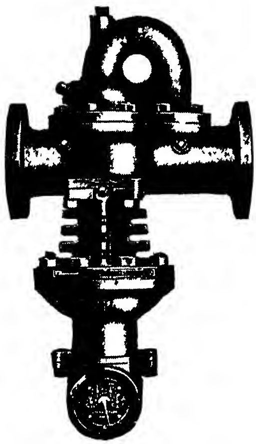
The VENTURI METER for water supply, boiler feed and other main pipe lines.
The CHRONOFLO ELECTRIC FLUID METER for long distance transmission of flow rates, quantities, pressures, temperatures, etc.
The SHUNT METER for steam, air and gas.



THE CHRONOFLO ELECTRIC FLUID METER

"From hundreds of feet to hundreds of miles"

The Chronoflo has introduced new features to the field of metering. Four common electric units are utilized: the synchronous motor; the mercury switch; the magnetic relay; the magnetic clutch. Electric impulses of variable time intervals proportional to rates or positions are transmitted by the primary unit to the receiver gauge. Regular A.C. current is used and accuracy is not affected by voltage variation. Suitable for intramural or long distance transmission over regular telephone channels.



Shunt Steam Meter, Type K.S.

THE SHUNT METER

Distinctly filling the need for a low priced, practical, *mechanical*, meter easily installed and accurate over a wide range. For measuring steam sold or in checking distribution of steam (also air or gas) to buildings, departments or processes. The illustration shows the complete meter. Installation consists simply of bolting to flanges in the flow line; no connecting pipe or electric wiring. A portion of the entering steam is deflected by an orifice through nozzles against the blades of a turbine located in the upper or shunt passageway. The speed of the turbine is kept low by a damping fan on the vertical turbine shaft, at the bottom end of which is a magnetic drive to the totalizer dials. The meter is installed as a unit in 2, 3 and 4-inch lines; for larger capacities the installation is made in a by-pass around an orifice in the main line.

	Steam Pressure Lbs. per Sq. In. Gauge	Rated Capacity of Saturated Steam—Lbs. per Hour (Meters with largest orifice)						
		2″ Meter	3″ Meter	4″ Meter	6″ Meter	8″ Meter	10″ Meter	12″ Meter
Low Pressure Meters	0	600	1,400	2,530	5,660	9,850	15,200	19,000
	5	800	1,900	3,350	6,500	11,300	17,500	21,800
	15	1200	2,750	4,750	7,860	13,700	21,100	26,300
	30	1650	3,400	5,850	9,550	16,600	25,600	31,900
	50	2000	4,050	7,000	11,400	19,800	30,500	38,000
Standard Meters	50	2650	6,000	10,600	22,200	39,000	60,500	75,500
	100	4150	9,500	16,600	29,000	51,000	79,200	99,000
	150	5000	11,400	20,000	34,400	60,500	94,000	117,000
	200	5700	13,000	22,700	39,000	68,600	107,000	133,000
	250	6300	14,400	25,200	43,200	76,000	118,000	148,000
Extra Heavy Meters	275	6600	15,200	26,500	45,000	79,400	123,000	154,000
	300	6900	15,700	27,500	46,800	82,500	128,000	160,000

Minimum Capacities equal one-tenth tabulated quantities. Many other intermediate capacities available.
Bulletin No. 255 gives complete information.

Century Electric Company

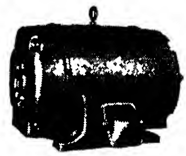
1806 Pine Street, **St. Louis, Mo.**

Offices and Stock Points in Principal Cities

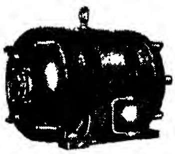
A MOTOR FOR EVERY PURPOSE

One Speed—Multispeed—Alternating and Direct Current

← 2 Horse Power Repulsion Start Single Phase Induction Motor. Sizes 1/8 to 40 horse power.



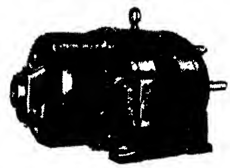
25 Horse Power Splash Proof → Squirrel Cage Induction Motor. Sizes 1/2 to 30 horse power.



← 5 Horse Power Squirrel Cage 3 Phase Induction Motor. Sizes 1/6 to 600 horse power.

1/6 Horse Power Capacitor → Single Phase Induction Motor. Sizes 1/8 to 5 horse power.

← 1/6 Horse Power Split Phase Induction Motor. Sizes 1/6 to 1/3 horse power.



30 Horse Power Slip Ring 3 → Phase Induction Motor. Sizes 1/4 to 250 horse power.

OTHER Century PRODUCTS

SINGLE PHASE MOTORS—1/250 to 40 Horse Power.
SQUIRREL CAGE MOTORS—1/6 to 600 Horse Power.
SLIP RING MOTORS—1/4 to 200 Horse Power.
DIRECT CURRENT MOTORS—1/20 to 150 Horse Power.

ALSO:

Multispeed Motors
Blower Motors
Unit Heater Motors
Capacitor Motors
Cushion Mounted Motors
Refrigeration Motors
Totally Enclosed Fan Cooled Motors
Enclosed Motors

Splash Proof Motors
Explosion Proof Motors
Gasoline Pump Motors
Portable Motors
Elevator Motors
Rotary Converters
Vertical Motors
Motor Generator Sets

GENERAL ELECTRIC COMPANY

SCHENECTADY, N. Y.

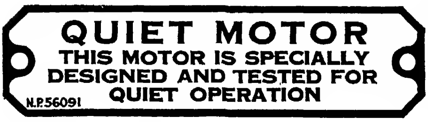
SALES OFFICES, WAREHOUSES, SERVICE SHOPS AND DISTRIBUTORS IN PRINCIPAL CITIES

For Code Wire, Conduit Products, Wiring Devices, Insulating Materials, etc., address Merchandise Department, Bridgeport, Conn.

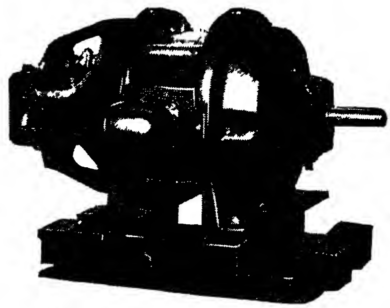
Quiet Operating Motors and Control

For Heating, Ventilating and Air Conditioning Systems

Modern buildings should be equipped with motors and other equipment especially designed for quiet operation. The General Electric Type KB and MB "quiet" motors are especially designed, built and tested for quiet operation and furnished with this nameplate:



Furthermore, the apparatus should be so installed that no vibrations are transmitted to the building structure as no matter how skillfully the equipment is designed and built, some vibrations (principally magnetic) must remain and if uncontrolled, they may appear as noise. The General Electric Company's "sound isolating base" is designed for use with General Electric motors to check the transmission to the building structure of vibrations which otherwise may appear as noise through amplification or resonance.

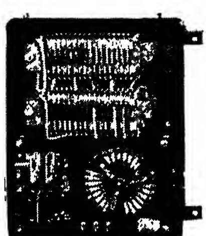


MB Wound-rotor quiet-operating motor mounted on sound-isolating base

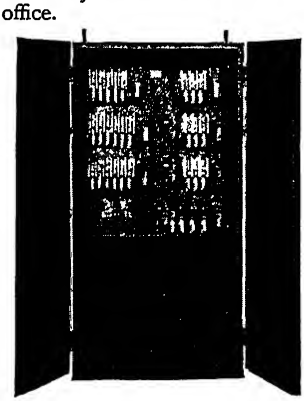
Alternating Current Control

The General Electric Co. manufactures a complete line of control devices for starting and controlling motors driving ventilating fans. All controllers are of the enclosed type externally operated. Speed regulators are designed to give 50 per cent speed reduction on fan load and arranged to maintain equalized current in all phases of the rotor which is necessary to maintain quiet operating motors.

Details may be obtained at the nearest sales office.



CR7765, 1-5 Hp. Controller



CR7761-F1, 20-50 Hp. Controller

FOR USE WITH WOUND-ROTOR MOTORS

This Company will gladly assist in the solution of any electrical problem in relation to heating and ventilation

The Lincoln Electric Company

13036 Coit Road, **Cleveland, Ohio**

Largest Manufacturers of Arc Welding Equipment in the World

Branch Offices and Distributors in All Principal Cities

Manual and Automatic Arc Welding Equipment, Electrodes and Supplies; A.C. Motors in Standard Types ½ to 200 H.P.

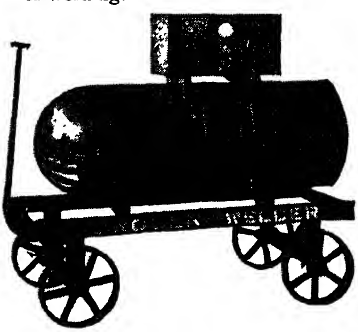
Shielded Arc Welding

Welds made by the shielded arc with Lincoln equipment have a tensile strength of 65,000 to 75,000 pounds per square inch, ductility and density equal to mild rolled steel and greater resistance to impact, fatifue and corrosion. Shielded arc welding is used in the fabrication of pipe systems, steam, water, oil and gas transmission lines, also in the fabrication of ducts and other sheet metal products, also in the construction of fan and blower housings, boilers, furnaces, tanks and pressure vessels. Shielded arc welding under the proper procedure satisfies all the requirements of the A.S.M.E. Boiler Code and insurance regulations.

The Lincoln "Shield-Arc" Welder

Lincoln "Shield-Arc" welders have many exclusive patented features which make possible the following 3-way guarantee of welding with "Shield-Arc" welders:

1. More weld metal deposit per K.W.H.
2. Faster welding per K.W.H.
3. Lower cost per unit of welding—the unit being per lineal foot of weld, or per pound of weld metal, or per hour of welding.



Lincoln "Shield-Arc" Welder

Lincoln welders, A.C. and D.C. motor driven, are built in 100 to 400 ampere sizes; belt and gasoline driven types in 100 to 600 ampere sizes.

Welding Electrodes

"Fleetweld"—For shielded arc welding of pipe, plate and shapes. Welding speed 150 to 300 per cent faster than ordinary welding.

"Lightweld"—For shielded arc welding of 16–24 gauge metal.

"Stainweld A"—For welding 18–8 stainless steel.

"Aluminweld"—For welding aluminum.

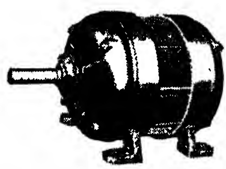
"Ferroweld"—The electrode that solves cast-iron welding problems.

"Hardweld"—A high carbon rod for hard surfacing.

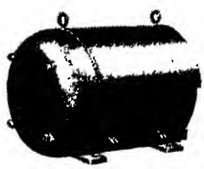
"Manganweld"—For welding high maganese steel.

"Stable-Arc"—A non-splashing rod for general welding purposes.

Lincoln "Linc-Weld" Motors



The Motor with the Extra Horsepower, Lincoln "Linc-Weld" Type D



Stainless Steel Motor, Lincoln "Linc-Weld" Type E, Totally Enclosed, Fan Cooled

Known as the motors which deliver extra horsepower without sacrifice of power factor or efficiency. Arc-welded rolled steel frames provide structural strength without bulk. This permits larger openings for greater ventilation, resulting in cooler operation of motors. Built in all standard types of polyphase induction motors for pump, fan and blower service, also for other general and special purposes.

Use of the stainless steel motor, the "Linc-Weld," Type E (totally enclosed, fan cooled), is recommended wherever there is dust, dirt, moisture or fumes in sufficient quantities to clog an open type motor or to abrade and corrode windings and bearings. Though completely sealed, the "Linc-Weld" Type E stainless steel motor operates continuously at full load well within the N. E. M. A. allowable temperature rise of 55 degrees Centigrade.

Westinghouse Electric & Manufacturing Co.

East Pittsburgh **Pennsylvania**

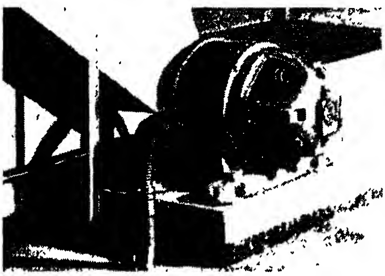
Research, Application and Design Engineers will cooperate and solve any electrical problem for heating and ventilating systems.

Sales Offices and Service Shops in all Principal Cities

Quiet Operating Motors

The Westinghouse Company has long recognized the need for quiet operating machinery and has devoted a vast amount of research work to the development of special instruments to measure noise accurately. The human element, which gives variable results, is eliminated in all testing procedure.

Westinghouse Quiet Operating Motors are specially designed and *individually*

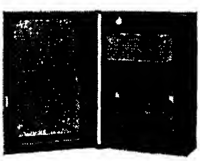


A Type CW Variable Speed, Quiet Operating Motor Driving Fan

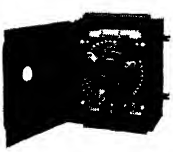
tested under loaded and unloaded conditions in a sound-proof room. A special nameplate is your assurance of their high standard of quiet operation.

Lasting quietness is assured by rigid, unit cast motor frames which contribute to the permanence of bearing alignment and air gap. Rotors are given a special dynamic balance, practically eliminating noises due to vibration.

Motor Control



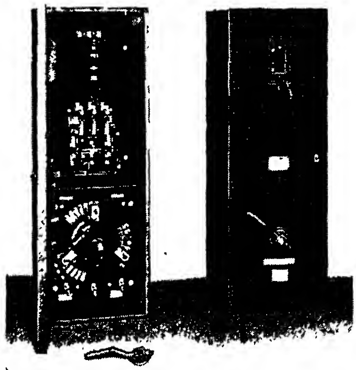
Linestarter for A-c. Drives



A Class 12-016 Speed Regulating Rheostat for use with Fan Drives

Starters, speed regulators, thermostats, switches and Nofuze circuit breakers are available for every requirement.

For fan drives, the new Westinghouse Type O-1 Controller offers unusual advantages. This unit assembly includes the equipment ordinarily housed in three separate cabinets. Primary Linestarter,



Open and Closed Views of Type O-1 Fan Drive Control, Wall-mounted Types

speed regulator and Nofuze breaker for circuit disconnect and protection are all mounted and wired in a neat cabinet for wall or floor mounting. Installation is greatly simplified and its appearance is improved.

Arc Welding

FlexArc A-c. Welder for Simplified Fabrication

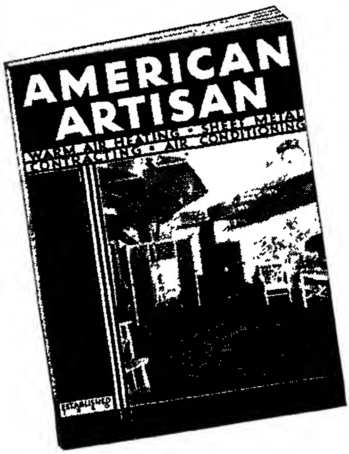
For the fabrication of thin-gauge materials such as ducts, Westinghouse offers the FlexArc A-C. welder. Its operation is extremely simple and it uses less than 15 cents worth of power an hour. One man can wheel it easily from place to place. It operates from the ordinary single-phase power circuit.

American Artisan

Published by

KEENEY PUBLISHING COMPANY

1900 Prairie Ave., **Chicago, Ill.**



AMERICAN ARTISAN, published monthly, serves the warm air heating, sheet metal contracting and air conditioning industry. It foresaw in the early stages of air conditioning and automatic heating development a field of great possibilities for its warm air heating dealer and sheet metal contractor readers, and for a number of years has given close editorial attention to the merchandising, installation and operation of this type of equipment and its accessories as developed for residential and similar applications.

Residential air conditioning and automatic heating have progressed rapidly and logically along the lines of the warm air heating type of system. To keep pace with the trends of the industry, AMERICAN ARTISAN devotes a special section in each issue exclusively to the technical and merchandising problems of air conditioning and automatic heating in homes and small buildings. Just as this field is an essential part of the warm air heating and sheet metal contracting industry, so is this section an essential part of AMERICAN ARTISAN.

Ventilation and forced air heating, together with all phases of sheet metal contracting and gravity warm air heating, make the entire magazine of interest and value to engineers and contractors who wish to keep up with this division of the heating, ventilating and air conditioning field.

AMERICAN ARTISAN is a member of the A. B. C. and A. B. P.

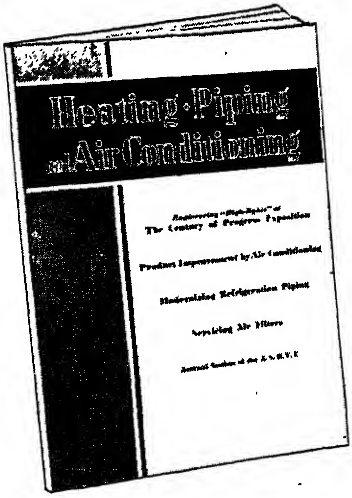
Subscription rates—$2.00 per year. *Advertising rates furnished upon request.*

Heating, Piping and Air Conditioning

Published by

KEENEY PUBLISHING COMPANY

1900 Prairie Ave., **Chicago, Ill.**



HEATING, PIPING AND AIR CONDITIONING is edited to give specialized attention to the design, installation, operation and maintenance of heating, piping and air conditioning systems in industrial plants and larger classes of construction.

It is a strictly technical journal, published monthly.

It carries in each issue, as a separate section, the journal of the American Society of Heating and Ventilating Engineers, and numbers among its subscribers the members of this Society.

Besides its regular staff of editors, fifteen consulting and contributing editors on heating, fourteen on piping and twelve on air conditioning insure a high grade, authoritative, technical and practical coverage of the subjects for which it is named.

HEATING, PIPING AND AIR CONDITIONING by concentrating editorially on these specialized services has singled out in the leading industrial plants of the country the one man whose major or sole concern is with this division of plant operation and maintenance.

Similarly, it has attracted the consulting engineers, the chief engineers of large buildings, and large contractors who require this specialized treatment of the services which constitute their interest and work.

Such a coverage means, for the advertiser, consideration at all points in the selling of a heating, piping or air conditioning product—consideration in the selection of a product during the preparation of plans and specifications; consideration in the actual purchase of a product for installation; consideration in the year 'round buying of a product for operating and maintenance requirements.

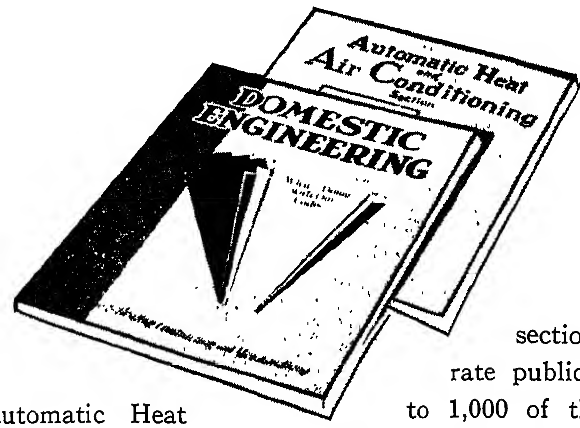
HEATING, PIPING AND AIR CONDITIONING is a member of the A.B.C. and A.B.P.

Subscription rates—$2.00 per year. *Advertising rates furnished upon request.*

Domestic Engineering

1900 Prairie Avenue

Chicago



THE Automatic Heat and Air Conditioning Section of DOMESTIC ENGINEERING has had an unusually interesting career. Starting in 1931 with a few articles, it developed into a department, and, as its importance increased, it grew into a special section.

Later, requests came from advertisers to send reprints of this section to lists of their special dealers, which included exclusive oil burner dealers, coal stoker dealers, dealers in air conditioning equipment, gas fired boiler dealers, and to other dealers outside of the regular heating trade.

The first distribution of this section as a separate publication went to 1,000 of these special dealers. Later other advertisers, realizing the importance of this separate distribution, which is in addition to the regular distribution as part of DOMESTIC ENGINEERING, added their lists of special dealers.

At the present time, in addition to its more than 15,000 circulation in DOMESTIC ENGINEERING, it goes to over 9,000 special dealers in automatic heating and air conditioning equipment every month, giving advertisers in the section blanket coverage of their market. This gives advertisers a blanket coverage of over 24,000 at the *lowest available cost per thousand.*

BURKS SUPER-TURBINE PUMPS *Only one moving part*

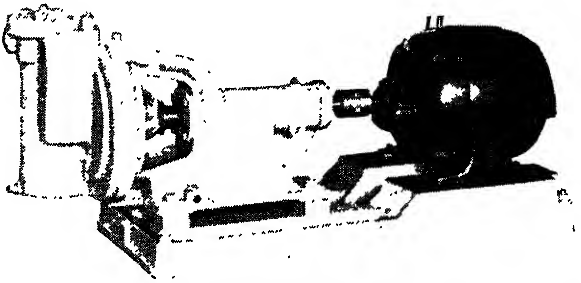
Decatur Pump Company

Decatur, Ill.

BURKS SUPER-TURBINE PUMPS *Only one moving part*

BURKS SUPER TURBINE PUMPS AND WATER SYSTEMS

Self Priming High Head Units



High Efficiency. Open-Impeller Type. Self Priming Centrifugal Pumps that are DIFFERENT!

Capacities to 400 G.P.M.

Built up to an Engineering Ideal.

Burks Self Priming Centrifugal Pumps

BURKS CONDENSATION RETURN UNITS

Will Not Steambind!

Ability to pump air alone, or air mixed with water, to the full pressure rating of the pump, means that the Burks pump will not steambind. Pump capacities range from 150 gph to 1,000 gph against boiler pressures up to 100 lbs. per square inch, and will take care of up to 12,000 sq. ft., of radiation per unit.

Hydraulically Balanced

These pumps are hydraulically balanced and cannot be made to pound or hammer under any condition of operation.

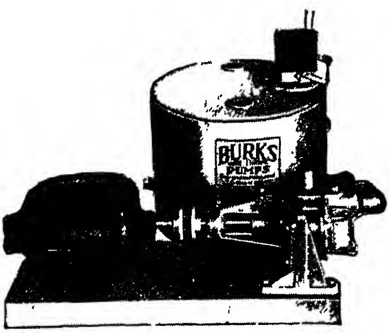
Construction Features

Only one moving part—the impeller—and this one part does not contact with metal at any stage of operation.

Self-priming and fully automatic. Impeller and raceway of cast bronze.

Shaft of stainless, non-corroding steel, impervious to mild acids, and carried in oversize ball bearings.

Tank of copper-bearing steel, 3/16-in. shell; 1/4-in. top and bottom, welded construction.



Series 41100
Condensation Return Unit

Series 4700
Condensation Return Unit

A SATISFACTORY CONDENSATION RETURN UNIT TO INSTALL

CHICAGO PUMP COMPANY

SEWAGE-CONDENSATION-CIRCULATING
BILGE-FIRE-HOUSE-VACUUM

2330 Wolfram Street - **BRUnswick 4110** - **Chicago**

PRODUCTS—Vacuum and Boiler Feed Pumps, Condensation, House, Booster, Fire Pumps, Circulating, Brine, Sewage, Bilge, Sludge, Pneumatic Tankless Water Systems and Automatic Alternator.

NRA MEMBER US WE DO OUR PART

Condensation Pump and Receiver

for Low, Medium and High Pressures
Systems up to 150,000 Sq. Ft. Radiation

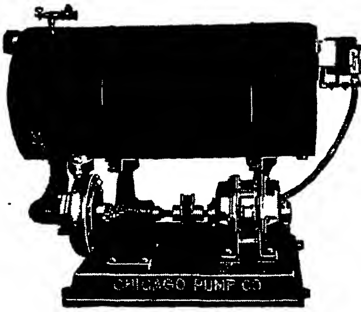


Fig. 1931—F. C Condensation Pump

"Chicago" Condensation Pumps are built for systems ranging from 2,000 up to 150,000 sq. ft. of radiation, and for boiler pressures up to 200 lbs. Units are built in either single or duplex—the duplex being alternated in their operation by the Automatic Alternator. *For tables and complete description ask for Bulletin 129.*

Vertical Condensation Pump

for Low and Medium Pressure for Systems up to 100,000 Sq. Ft. Radiation

Fig. 1940 Vertical Condensation Pump

The vertical condensation pump is designed to receive returns from lowest radiation. The receiver is placed underground—an ordinary hole sufficing if necessary — and requires very little floor space. Unit is shipped complete, easy to install, assembled so as to prevent steam leaks. Special bearings will stand up under hot water for several years. A special float mechanism is guaranteed not to leak or stick in stuffing box. *Complete data and description in Bulletin 133.*

"Sure-Return" Condensation Pump

for Low and Medium Pressure, and Systems up to 35,000 Sq. Ft. Radiation

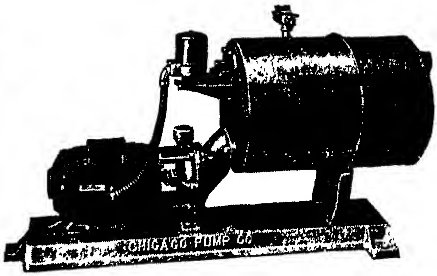


Fig. 1946

"Sure Return" Condensation Pumps and Receivers are built for systems up to 35,000 sq. ft. of direct radiation and for low and medium pressures. Built in either single or duplex units. Duplex units are alternated in their operation by the Automatic Alternator. *Complete data in Bulletin 131.*

Horizontally Split Pumps

for all Services

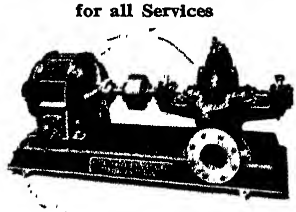


Fig. 1881—Single Stage Type "D" Pump

For any service (such as boiler feed, water supply, tank filling, circulating, fire protection, etc.). Chicago Pump Co. builds a line of horizontally split case centrifugal pumps in both single and multistages. Completely bronze fitted (except where special fittings are required) ball bearings, internal water seal, oil is filtered. "Chicago" Horizontal Pumps are built for efficient performance and long life.

"CONDO-VAC"

Return Line Vacuum and Boiler Feed Pump

Automatic Alternator is available on Duplex Return Line Vacuum and Boiler Feed Pumps

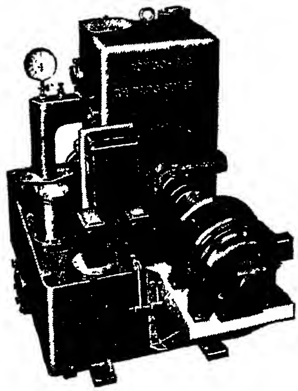


Fig. 2100—Single "Condo-Vac"

No vacuum on stuffing boxes, ample clearance in rotating member. It costs less to operate a Condo-Vac. Condo-Vac reduces corrosion in piping and boiler to minimum—because pump does not take in air from atmosphere and entirely eliminates all air coming back from system. Condo-Vac is quiet, has a low inlet, entirely automatic, fool-proof, easy to maintain. *Ask for Bulletin 137*, learn more about the modern vacuum pump with the long life principle of operation.

"Sure Return" Engineering Table

Unit No.	Maximum square feet direct radiation	Pounds pressure pump will discharge against	Horse power motor furnished	Size of discharge in inches	Size of return inlet in inches	Capacity of pump in gallons per minute	Capacity of receiver in gallons	Height of inlet in receiver from floor level in inches	Approximate shipping weight—Single
*SR602	2,000	10	1/4	1	2	3	16	21½	300
SR604	"	27	1/2	1	"	"	"	"	350
*SR606	"	50	1/4	3/4	"	"	"	"	400
*SR609	4,000	10	1/4	1	2	6	16	21½	300
SR611	"	21	1/2	1	"	"	"	"	350
*SR613	"	50	1/4	3/4	"	"	"	"	400
*SR616	6,000	10	1/4	1	2	9	16	21½	300
SR618	"	18	1/2	1	"	"	"	"	350
*SR620	"	50	1/3	3/4	"	7½	"	"	425
*SR623	8,000	10	1/4	1	2	12	19	23	300
SR625	"	17	1/2	1	"	"	"	"	350
SR627	"	17	3/4	1¼	"	"	"	"	375
SR629	"	22	1	"	"	"	"	"	400
SR631	"	29	1½	"	"	"	"	"	500
SR633	"	35	2	"	"	"	"	"	525
*SR636	10,000	10	1/4	1	2	15	19	23	350
SR638	"	15	1/2	1	"	"	"	"	400
SR640	"	16	3/4	1¼	"	"	"	"	425
SR642	"	21	1	"	"	"	"	"	450
SR644	"	29	1½	"	"	"	"	"	550
SR646	"	35	2	"	"	"	"	"	575
SR651	15,000	10	1/2	1¼	3	21	59	30	450
SR653	"	17	3/4	"	"	"	"	"	510
SR655	"	20	1	"	"	"	"	"	550
SR657	"	28	1½	"	"	"	"	"	650
SR659	"	34	2	"	"	"	"	"	675
SR662	20,000	10	1/2	1¼	3	30	59	30	475
SR664	"	16	3/4	"	"	"	"	"	510
SR666	"	19	1	"	"	"	"	"	550
SR668	"	26	1½	"	"	"	"	"	650
SR670	"	33	2	"	"	"	"	"	675
SR673	25,000	10	1/2	1¼	3	35	59	30	475
SR675	"	15	3/4	"	"	"	"	"	510
SR677	"	18	1	"	"	"	"	"	550
SR679	"	25	1½	"	"	"	"	"	650
SR681	"	30	2	"	"	"	"	"	675
SR684	35,000	12	3/4	1	3	50	59	30	510
SR686	"	15	1	"	"	"	"	"	550
SR688	"	22	1½	"	"	"	"	"	650
SR689	"	27	2	"	"	"	"	"	675

*Units marked with asterisk run at 1750 R.P.M. All other units listed on this sheet run at 3500 R.P.M.

"Condo-Vac" Engineering Table

Single Standard Capacity, with Double Automatic Control and Receiver. 20 Pounds Discharge Pressure

Square Feet Direct Radiation	Horsepower of Motor Furnished	Capacity for Water only, at 160° F. GPM	Simultaneous Water and Saturated Air Capacity at 160° F		Pump Discharge in Inches	Approximate Shipping Weight in Pounds
			Water GPM	Air CFM at 5½"		
*2,500	½	3.8	1.3	1.3	1¼	500
*5,000	¾	7.5	2.5	2.5	"	525
*10,000	1	15.0	5.0	4.0	"	550
15,000	1½	22.5	7.5	5.4	"	1,000
20,000	2	30.0	10.0	6.8	1½	1,200
25,000	2	37.5	12.5	8.3	"	1,300
30,000	3	45.0	15.0	9.7	"	1,450
40,000	5	60.0	20.0	12.6	2	1,700
65,000	7½	97.5	32.5	19.8	"	2,200
100,000	10	150.0	50.0	30.0	3	3,000
150,000	15	225.0	75.0	50.0	"	4,000

*Vertical Style Condo-Vac (Fig. 2120) operating at 3500 R.P.M.

Goulds Pumps, Inc.

Seneca Falls, New York

New York.......16 Murray St. Tulsa.......213 E. Archer St. Pittsburgh.......636 H. W. Oliver Bldg.
Philadelphia.......111 N. Third St. Chicago....53 W. Jackson Bl.d. Atlanta.......22 Marietta St. Bldg.
Boston.......8 Albany St. Houston.......P. O. Box 965

Manufacturers of Pumps for Every Service

PRODUCTS—Centrifugal Pumps for all purposes—Single and Double Suction, Sump, Fire, Single and Multistage, Horizontal and Vertical. Triplex Pumps.

Single and Double Acting, Deep Well, Power Rotary, Pressure Pumps, etc.

Goulds Pumps, manufactured since 1848, comprise a line of hand and power pumps, including types and sizes for practically all pumping services.

Every Goulds Pump sold is guaranteed to give reliable, satisfactory service under the conditions for which it is recommended.

The power pumps can be furnished for belt, chain, gear or direct drive from all types of drivers.

Condensation Return Pump and Receiver

For handling returns which come back at or below the floor line of boiler room. Will handle returns from 2,000 to 25,000 sq. ft. of radiation; good for pressures up to 28 pounds. Outfits shipped from stock completely assembled.

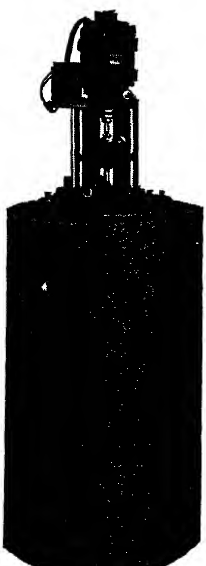


Fig. 3354

Pump—Ball bearing centrifugal type, two stage, bronze fitted. Requires no lubrication.

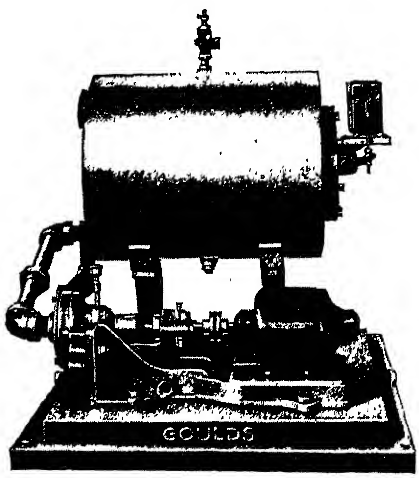
Pump Shaft and Float Rod—Operate through stuffing boxes to prevent leaks. This enables outfit to be used under pressure. Float Rod cannot bind.

Tank—Heavy, galvanized, welded steel, 24 in. in diameter by 40 in. high. Tank has two inlets both tapped for 3 in. pipe, one 8 in. the other 14 in. below top of tank. Tapped for vent connection, and fitted with drain opening. 1-in. discharge from pump to boiler.

Motor—1750 r.p.m., 110-220 volt, 60 cycle repulsion induction motor.

Fig. 3354 has ½ hp. motor; Fig. 3356 has 1 hp. motor.

Capacity—Pumps are suitable for Maximum Direct Radiation, 25,000 sq. ft.; Maximum Discharge Pressure, 28 lb.; Height Over-all, 64⅜ in.; Weight 350 lb.



Horizontal Condensate Return Receiver

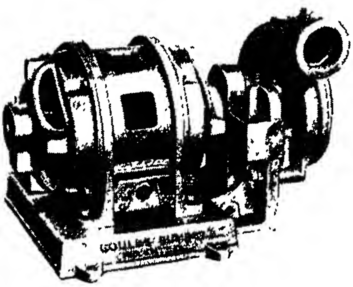
The unit consists of a horizontal, galvanized, corrosion-resistant, welded steel receiver tank, in 20, 40 or 60 gallon capacity, with float and automatic switch mounted upon bedplate as illustrated.

Pump is the Goulds well known Flexi-Unit type with capacities from 5 to 60 g.p.m. Pressures up to 20 lbs. Complete units can be furnished for radiations up to 40,000 sq. ft.

For higher ratings, special units can be furnished.

Close Coupled Centrifugal Pumping Units

A compact, low cost and efficient centrifugal pumping unit with a capacity range up to 1000 g.p.m. Heads range from 10 feet up to 290 feet. Pumps are built in sizes $\frac{3}{4}$ in. to 4 in. For general pumping service in industrial plants, offices and apartment buildings, green-houses, refineries, cold storage plants—in fact, its application is universal. For any installation where an inexpensive yet efficient and dependable unit is required, a Goulds Close-Coupled unit will give years of trouble-free service.



Pumps can be furnished in standard fitted, all iron, bronze fitted or all bronze construction. Standard motors without shaft extensions may be used if they have the proper ball bearings. No special end plates or brackets required. Short bed-plate provides positive support for pump independent of motor. Self venting top horizontal discharge gives quiet running pump with no loss in capacity or head due to air binding.

Write for Goulds Bulletin 204 for complete description and performance chart.

Automatic Cellar Drainers

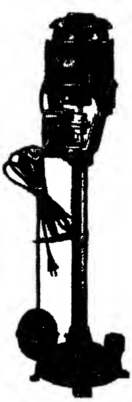


Fig. 3108

Self-Contained Vertical Type—Fig. 3108. Capacities up to 40 gals. per minute. Heads up to 25 feet. For use in an 18-inch diameter tile or ordinary barrel set in a pit 2 feet deep.

Shipping weight, 75 lbs.

Pump has cast bronze casing, bronze impeller and stainless steel shaft. Lower bearing made of non-scoring, non-seizing, self-lubricating, "Bearium" metal. Upper bearing is combined radial and thrust type ball bearing.

Motor is $\frac{1}{4}$ hp., for 25 or 60 cycle, A.C. or D.C. circuits.

Float switch is entirely enclosed in housing which is part of motor support. Float is heavy gauge copper with brass rod and adjustable stops.

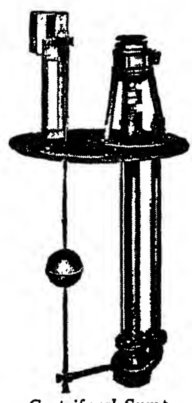
Complete with 10 feet of rubber covered cord and plug.

Horizontal Self Priming Type—Fig. 3034. Two sizes, capacities 1 in. size up to 30 gal. per minute; 1$\frac{1}{2}$ in. size up to 50 gal. per minute. Heads up to 20 ft. Approximate weight, 100 and 250 lb.

This type is equipped with special priming chamber making it entirely automatic for pit depths of 3$\frac{1}{2}$ ft. Minimum diameter drainage pit, 1 in. size, 13$\frac{1}{2}$ in.; 1$\frac{1}{2}$ in. size, 16 in.

Pump is bronze fitted. Discharge 1 in. or 1$\frac{1}{2}$ in. Pump shaft is connected to motor shaft by flexible coupling. Motor is repulsion induction type. Double pole type switch. Copper float with brass rod.

Centrifugal Sump Pump—Fig. 3047. Electrically driven, directly connected to motor. Single stage, single suction. Used to elevate drainage in buildings to street level, where basement floors, boilers and elevator pits are below sewer level, and for any other service where liquid accumulates in a catch basin, pit or tank. Furnished with non-clogging bronze impeller to prevent corrosion during idle periods. Outfit includes pump, motor and control completely assembled and shipped from stock. Built in 1$\frac{1}{2}$, 2, 3, and 4 inch sizes for capacities up to 650 G. P. M.; heads up to 70 feet. Motors $\frac{1}{2}$, $\frac{3}{4}$, 1, 1$\frac{1}{2}$, 2, 3, 5, and 7$\frac{1}{2}$ carried in stock with automatic control. All pumps may be furnished for pit depths up to 14 ft. Duplex units also available.

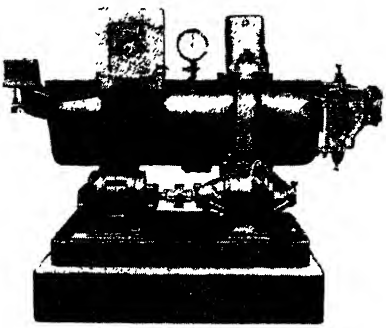


Centrifugal Sump Pump Fig. 3047

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities



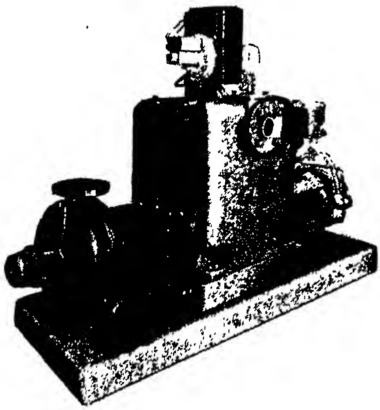
Jennings Return Line Vacuum Heating Pumps

Standard with the heating industry for over sixteen years. They remove air and condensation from the return lines of vacuum steam heating systems, discharging the air to atmosphere and returning the water to the boiler.

Two independent units are combined in a single casing—an air unit and a water unit. Impellers of both are mounted on the same shaft. The pump is bronze fitted throughout.

Supplied either direct connected to standard electric motors, for belt drive, or for steam turbine drive. For continuous or automatic operation against pressures up to 40 lbs. Supplied standard in capacities up to 300,000 sq. ft. E.D.R.

Complete data in Bulletin No. 85 on request.

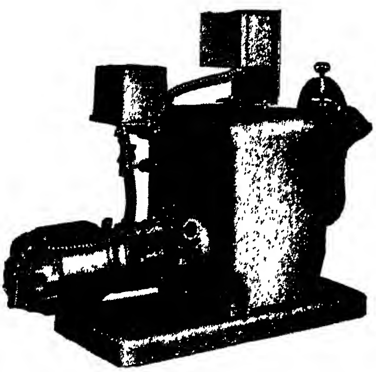


Jennings Vapor Turbine Vacuum Heating Pumps

The Jennings Vapor Turbine Heating Pump combines all of the advantages of the standard return line heating pumps with a new type of drive, a specially designed low pressure turbine which operates directly on steam from the heating mains on any system, requiring a differential of only 5 in. of mercury, and returns that steam to the heating system with practically no heat loss.

This pump affords the safety and economy which goes with a continuous condensation return and steady vacuum, and at no cost for electric current. Furnished standard in capacities up to 30,000 sq. ft. E.D.R.

Complete data in Bulletin No. 203 on request.



Condensation Pump and Receiver

Removes condensation from radiators in return line steam heating systems and pumps condensation back to the boiler.

They are sturdy and compact in construction, and combine receiving tank, pump and driving motor in a single assembly. Bronze fitted throughout, with Tobin bronze shaft. Impeller is of special design adapted to handling hot water with highest efficiency.

Jennings Condensation Pumps are furnished in standard sizes with capacities ranging from 4 to 200 g.p.m. of water. For serving up to 150,000 sq. ft. of equivalent direct radiation.

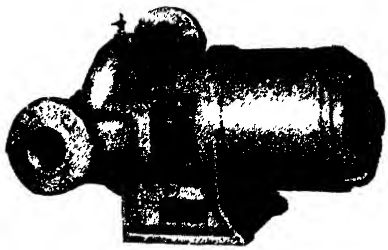
Complete data in Bulletin No. 165 on request.

The Nash Engineering Company

South Norwalk, Conn., U. S. A.

Sales and Service Offices in all Principal Cities

Centrifugal Pump



Made in standard and suction (self-priming) types. For circulating hot and cold water; boosting city water pressure; handling water in air washing and conditioning; handling ash sluicing water, etc.

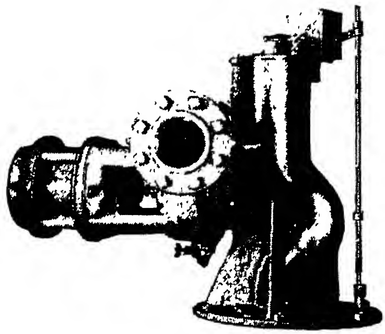
Compact—motor armature and pump impeller are mounted on the same shaft. Simplified—no bearings in pump casing, one stuffing box. Accessible—impeller removable without disturbing piping or shaft alignment.

Self-priming types will handle air or gas continuously with liquid being pumped, and can be operated intermittently without foot valve.

Supplied in 1¼, 1½, 2, 3, 4, 6, and 8 in. sizes with capacity up to 2000 g.p.m. Heads up to 300 ft.

Complete data in Bulletins 155, 159 and 161, on request.

Suction Sump and Sewage Pumps



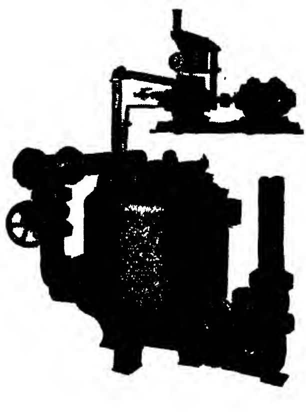
Jennings Suction Sump Pumps are self-priming centrifugals for handling seepage water and liquids reasonably free from solids. The Suction Sewage Pumps are equipped with a non-clog type impeller for liquids containing solids. Suction piping only is submerged. Centrifugal impeller and vacuum priming rotor are both mounted on same shaft that carries rotor of the driving motor, forming a single moving element and rotating without metallic contact.

These pumps will handle air or gas with liquid being pumped, and because of self-priming feature are installed entirely outside of pit. This affords perfect accessibility for inspection or cleaning.

Capacities to meet all requirements.

Complete data in Bulletins 159 and 161, on request.

Sewage Ejector



For pumping unscreened sewage or drainage from basements below street sewer level; handling crude sewage from low level districts; pumping effluent, sludge and other heavy liquids.

The Jennings Sewage Ejector is of exclusive pneumatic design, no sewage passing through the pump. Air is furnished by the Hytor Compressor at operating pressure, and compressor operates only when air is required.

Reciprocating compressors, air valves, high pressure air storage tanks, and pressure reducing valves have all been eliminated.

Furnished in standard sizes for handling from 30 to 1500 g.p.m. Heads up to 50 ft.

Complete data in Bulletins 103 and 108, on request.

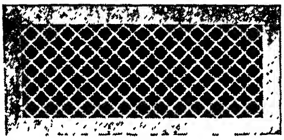
Hart & Cooley Manufacturing Co.

General Sales Office: 61 W. Kinzie St., **Chicago**

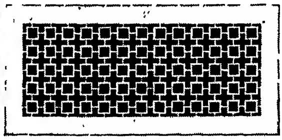
PRODUCTS—Steel, Semi-Steel and All Cast Registers; Steel and Cast Register Faces and Reinforced Cold Air Face Plates; Ceiling and Sidewall Ventilators; Lock Registers; Heat Control; Damper Regulators; Chain; Pulleys; Furnace Accessories.

A COMPLETE LINE OF FORCED-AIR REGISTERS

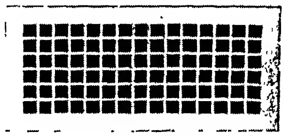
With Choice of Three Designs Shown Below



No. 30 Grille Design



No. 40 Grille Design



No. 50 Grille Design

A TYPE FOR EVERY CONDITION

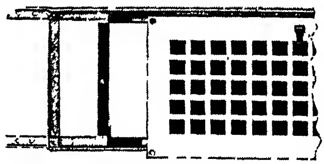
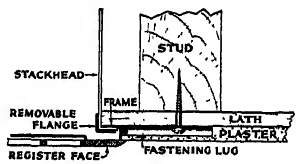


Diagram below shows Installation of 3-Piece Register



3-Piece Sidewall Register—Consists of a removable grille face, plaster frame, and detachable flange. Frame has extension arms which are fastened directly to studs, eliminating all special carpentry work and resulting in a rigid installation. Flange holds stackhead permanently in place and prevents streaking. Face overlaps edge of frame, effectively concealing plaster joint.

1-Piece Sidewall Register—Consists of a grille face and detachable flange. Stackhead is formed over flange and face is drawn up to wall by means of screws engaging in flange. Ideal for either old or new house work.

3-Piece Baseboard Register—Similar in construction to 3-piece sidewall register except frame has a depth of ⅞ in. Detachable flange results in a permanently streak-proof installation.

1-Piece Baseboard Register—Consists of an integral face and frame to which is welded a flange for engaging stackhead. Easy to install and especially suitable for old house work.

Return Air Intakes—Flanged Intake—Has a depth all around of ⅞ in. Suitable for use where intake extends above top of baseboard.

Flat Intake—Designed for installations in which the baseboard height is greater than that of the intake.

Advantages of Flat Type Grille

1. Easily decorated to match interior, resulting in inconspicuous installation. **2.** Does not collect dust. **3.** Easily cleaned or redecorated. **4.** Strong and rigid—not easily dented. **5.** Will not rust. **6.** Offers ideal control over velocity of air entering room. **7.** Causes air to enter room on a nearly horizontal plane. **8.** Does not cause any audible noise.

CLASS NUMBERS

Type Register	Design No. 30	Design No. 40	Design No. 50
3-Piece Sidewall	3331	3341	3351
1-Piece Sidewall	3330	3340	3350
3-Piece Baseboard	3131	3141	3151
1-Piece Baseboard	3130	3140	3150
Flanged Intake	3630	3640	3650
Flat Intake	3635	3645	3655

STANDARD SIZES

Registers	Flanged Intakes	Flat Intakes	
10 x 4, 5, 6, 8	12 x 4, 5, 6	12 x 4, 5	24 x 4, 5
12 x 4, 5, 6, 8, 9, 10	14 x 4, 5, 6	14 x 4, 5	30 x 4, 5
14 x 4, 5, 6, 8, 10	24 x 4, 5, 6	18 x 4, 5	
30 x 4, 5, 6, 8	30 x 4, 5, 6, 8	20 x 4, 5,	

The above registers are stocked in all leading finishes. Special sizes and finishes furnished promptly.

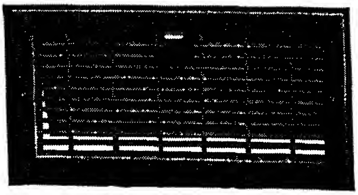
Send for Complete Catalog with Chart Showing Proper Sizes for All Conditions

Independent Register & Mfg. Co.

ESTABLISHED 1898

3753 East 93rd Street, **Cleveland, Ohio**

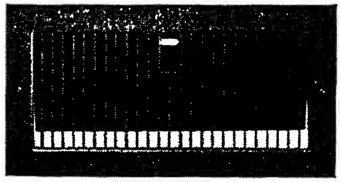
"Fabrikated" and Wrought Steel Registers and Grilles



Design No. 311—Single Valve

Two of 25 Designs to Choose From

Wall Registers

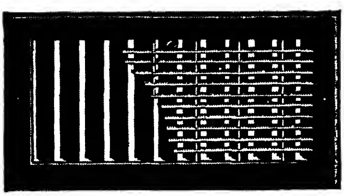


Design No. 211—Single Valve

Nos. 311, 300, 211, 201, 175-12SV With Web Wall Frames (Styles WX and BX)

SIZES AND LIST PRICES

To Fit Stackhead Size: (Horizontal Dimension First) Inches	"Fabrikated" Daylight Opening Size Inches	"Fabrikated" Overall Size Straight Edge Faces. Bev. Edge Faces 1/16" less	Black Japanned or Prime Coat	White Japanned Oak, or Lacquered Finishes	Electroplated	
					Oxidized Copper	Chrom., Nickel, Brass, or Bronze
10 x 4	9¼ x 2⅞	11⅜ x 5	$2.15	$2.50	$3.35	$3.90
10 x 5	9¼ x 3⅞	11⅜ x 6	2.30	2.65	3.60	4.20
10 x 6	9¼ x 4⅞	11⅜ x 7	2.40	2.80	3.80	4.40
10 x 8	9¼ x 6⅞	11⅜ x 9	2.50	3.00	4.20	5.00
12 x 4	11¼ x 2⅞	13⅜ x 5	2.30	2.65	3.60	4.15
12 x 5	11¼ x 3⅞	13⅜ x 6	2.40	2.80	3.80	4.35
12 x 6	11¼ x 4⅞	13⅜ x 7	2.55	2.95	4.05	4.65
12 x 8	11¼ x 6⅞	13⅜ x 9	2.65	3.30	4.55	5.05
12 x 9	11¼ x 7⅞	13⅞ x 10	2.90	3.45	4.75	5.35
12 x 10	11¼ x 8⅞	13⅜ x 11	3.10	3.60	5.00	5.55
14 x 4	13¼ x 2⅞	15⅜ x 5	2.75	3.20	4.30	4.95
14 x 5	13¼ x 3⅞	15⅜ x 6	2.80	3.25	4.35	5.05
14 x 6	13¼ x 4⅞	15⅜ x 7	2.85	3.35	4.45	5.15
14 x 8	13¼ x 6⅞	15⅜ x 9	3.00	3.50	4.70	5.45
14 x 10	13¼ x 8⅞	15⅜ x 11	3.15	3.70	4.95	5.75
16 x 4	15¼ x 2⅞	17⅜ x 5	2.95	3.45	4.65	5.35
16 x 5	15¼ x 3⅞	17⅜ x 6	3.05	3.55	4.75	5.50
16 x 6	15¼ x 4⅞	17⅜ x 7	3.10	3.65	4.85	5.65
16 x 8	15¼ x 6⅞	17⅜ x 9	3.35	3.90	5.25	6.05
16 x 10	15¼ x 8⅞	17⅜ x 11	3.60	4.15	5.65	6.45
20 x 4	19¼ x 2⅞	21⅜ x 5	3.40	4.00	5.30	6.10
20 x 5	19¼ x 3⅞	21⅜ x 6	3.55	4.15	5.50	6.35
20 x 6	19¼ x 4⅞	21⅜ x 7	3.65	4.30	5.70	6.60
20 x 8	19¼ x 6⅞	21⅜ x 9	4.00	4.70	6.30	7.30
20 x 10	19¼ x 8⅞	21⅜ x 11	4.35	5.10	6.90	8.00
24 x 4	23¼ x 2⅞	25⅜ x 5	3.85	4.45	6.00	6.95
24 x 5	23¼ x 3⅞	25⅜ x 6	4.00	4.65	6.25	7.30
24 x 6	23¼ x 4⅞	25⅜ x 7	4.20	4.90	6.60	7.65
24 x 8	23¼ x 6⅞	25⅜ x 9	4.65	5.45	7.35	8.55
24 x 10	23¼ x 8⅞	25⅜ x 11	5.10	6.00	8.10	9.45
30 x 4	29¼ x 2⅞	31⅜ x 5	4.50	5.20	7.00	8.10
30 x 5	29¼ x 3⅞	31⅜ x 6	4.75	5.50	7.40	8.60
30 x 6	29¼ x 4⅞	31⅜ x 7	5.00	5.80	7.85	9.10
30 x 8	29¼ x 6⅞	31⅜ x 9	5.65	6.60	8.95	10.40
30 x 10	29¼ x 8⅞	31⅜ x 11	6.30	7.40	10.05	11.70



Multiple Valve Registers—For Wall, Baseboard or Ceiling. Valves swing from their edge, when closed lay flat on back of register. The frame is ⅞" deep, not cutting down duct capacity. Moderate additional cost over single valve types.

Ask for Complete Forced Air Register Catalogue

Tuttle & Bailey, Inc.

HART & COOLEY **WM. HIGHTON & SONS**

New Britain, Conn.

Branch Offices: New York, N. Y. Boston, Mass. Philadelphia, Pa. Chicago, Ill.

PRODUCTS—Ornamental Grilles, Ventilating and Air Conditioning Registers, Convection Heaters, Cast Bronze Tablets.

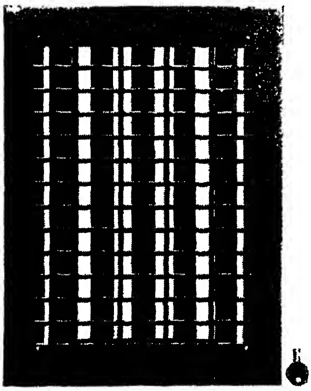
IMPORTANT NOTICE

On April 1, 1933, the Hart & Cooley interests acquired the goodwill of the Tuttle & Bailey Mfg. Co. Hart & Cooley, Tuttle & Bailey, and Wm. Highton & Sons are all combined into one company, known as "Tuttle & Bailey, Inc." Increased facilities and the centralization of efforts will give greater benefits and better service to the Engineering profession as well as to the trade in general.

McKNIGHT REGISTER, No. 1000

McKnight Patent Pending

FOR HEATING, VENTILATING, AND AIR CONDITIONING



No. 1000—Key Operated

The McKnight Register, No. 1000, was designed by a practical Engineer who had in mind the problems involved in laying out and operating a ventilating system. This register gives positive volume control, a vital feature in a register used in air conditioning systems.

Various tests that have been made on this register prove conclusively that the volume of air may be controlled from 100 per cent of normal down to nothing by simply turning a key at the register face, provided a system is designed for equal resistance.

A resistance is set up by this register at the outlet with the result that air delivered to the register face is less subject to changing pressures from the outside. Furthermore, an even distribution of the air flow over the entire register face is obtained without the necessity of using diffusers; this permits an easy and correct determination of the air velocity by means of an anemometer.

Outstanding Advantages of the McKnight Register

1. Positive Control of air volume with minimum adjustment.
2. Ease of balancing system.
3. No need of other dampers or any diffusers.
4. No counter effects in system caused by open windows or doors.
5. Minimum of air leakage through register.
6. Register has the appearance of a grille.
7. Air flow always uniform and at right angles to the register face.
8. No whistling noises perceptible.
9. Positive operation by key control.
10. Minimum resistance to air flow with register full open.
11. More accurately balanced system.
12. System costs no more than if ordinary registers are used.

A List of McKnight Register Installations will be gladly sent to Engineers on request

Uni-flo Grille Corporation

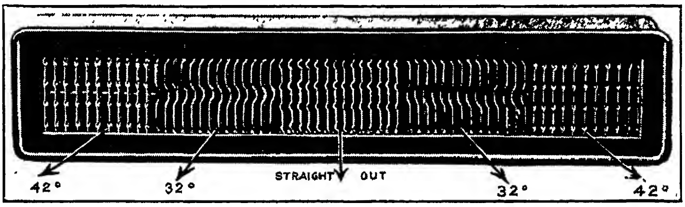
4646 Lawton Avenue **Detroit, Michigan**

uni-flo

Counselors of Air Distribution

PRODUCTS—Directional Air Distribution Outlets
for
Heating, Ventilating, Cooling and Air Conditioning Systems

Literature Upon Request



Uni-flo 5-Way
AIR DIRECTIONAL-OUTLET

Construction—
The construction of UNI-FLO Products—are of the fin and bar design.

Performance—
UNI-FLO-Directional Air Flow Outlets direct the air to almost any desired position by using combinations of different type cores. Air may be controlled to flow right, left, fan shape, or in up or down direction. When air flows through UNI-FLO Outlets it is separated into ribbon-like thin sheets and by the aid of diffusers formed on the edges of the fins, the air is broken up at the face of the outlet in such a way that a rolling action to the air is created causing an asperating effect at the face of the outlet that induces room air to mix with the discharged air thus tempering the air at the face of the outlet before its distribution throughout the room, this is a very desirable feature especially when refrigerated air is distributed as the pre-tempering of the air at the outlet increases the distance of air delivery at lower velocities insuring quiet air discharge.

Air-Noise—
UNI-FLO-Discharge Outlets are exceptionally quiet up to 1400 ft. only registering 35 decibels of noise at this velocity.

Refrigerated Air—
Air that is 15 degrees lower than room temperature will travel approximately 45 feet when discharged through UNI-FLO Outlets at a velocity of 1400 ft. per min.

Air-Control—
Individual controlled dampers are provided either key operated or spring actuated so that all or part of the air flowing from the outlet can be controlled. A very important feature where control of temperatures have-to-be-maintained-regardless-of-the-number of-people-in-the-room.

Sizes—
In inches from 4 in. to 96 in. long and 4 in. to 48 in. high.

Republic Steel Corporation

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TONCAN COPPER MOLYBDENUM IRON

What Is Toncan Iron?

Toncan Iron is a ferrous alloy of scientifically refined iron, copper and molybdenum in correct proportions—the alloy which ranks first in rust-resistance, among the ferrous metals, after the stainless alloys.

Thousands of rigid tests and the performance of untold tons of Toncan Iron in actual service point to the greater economy in the heating and ventilating systems of which the sheet metal and pipe are Toncan Iron.

Advantage of Toncan Iron

(1) It resists to a higher degree more of the many and varied types of corrosion than any other ferrous material except the stainless alloys. This resistance is not confined to the surface or "skin" of the metal, but is uniform throughout.

(2) It combines the high rust-resistance of an alloy iron with many of the desirable physical qualities of less resistant ferrous materials.

(3) It is one of the most workable of materials. Sheets form easily. Pipe may be handled like any iron.

(4) Unlike other ferrous materials, cold working—cutting, bending, punching, threading, etc.—in no way affects the rust-resistance of Toncan Iron.

(5) It welds easily by any of the usually accepted modern methods. The use of Toncan Iron welding rod insures an installation of equal rust and corrosion-resistance throughout.

(6) A uniform and tightly adherent galvanized coating can be applied, thus adding the protection of a heavy coating of zinc to the already high rust-resistance of the base metal itself.

(7) Through its longer, trouble-free life, it has been found to cost far less per year of service. Its use is more than an economy. It is insurance against sheet and pipe failures and costly replacements.

Toncan Iron Sheets

Toncan Iron is available in various sheet forms. Plain sheets may be had black or galvanized, in gauges No. 8 to 28, widths from 24 to 50 inches, and lengths from 10 to 13 feet, depending upon gauge and width. The Toncan Iron trade mark is stencilled in two or three places on every sheet.

Toncan Iron Pipe

Toncan Iron Pipe is available in standard, extra strong and double extra strong; black or galvanized; in sizes from 14-inch to 16-inch O.D. All Toncan Iron Pipe, 2-inch and larger, is electric resistance welded, and combines the foregoing advantages of Toncan Iron with the advantage of Republic's electric welding process—100 per cent weld, perfect roundness, uniform diameter and wall thickness, and freedom from scale. Toncan Iron Pipe, black, is painted blue; galvanized finish is marked with two blue stripes. Couplings are stamped RT.

Source of Supply

Toncan Iron Sheets and Pipe are stocked by jobbers in all large cities; leading contractors everywhere use Toncan Iron and are glad to supply it where specified. If, for any reason, you cannot obtain Toncan Iron, write to Republic's nearest sales office.

Literature

Two 64-page books, "The Path to Permanence" on Toncan Iron Sheets and "Pipe for Permanence" on Toncan Iron Pipe, will bring you the complete story. Write for your copies.

Other Republic Products

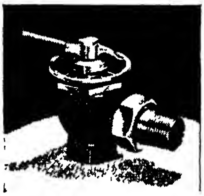
Republic Steel Corporation manufactures hundreds of iron, steel and alloy products, among which of interest to heating and ventilating engineers are Enduro Stainless Steel in sheets and other usual forms, steel pipe, and steel or Toncan Iron boiler tubes.

Barnes & Jones
INCORPORATED

129 Brookside Avenue, Jamaica Plain, Boston, Mass.

New York Office: 101 Park Avenue

Barnes & Jones Vapor and Vacuum Systems of Steam Heating; Radiator Valves; Metering Orifice Supply Valves; Thermostatic Radiator Traps; Thermostatic Traps for medium and high pressure; Condensators (Boiler Return Traps); Blast Traps; Drip Traps; Vent Traps; Strainers; Damper Regulators; Gages; Proportionator Systems with Zone Control.

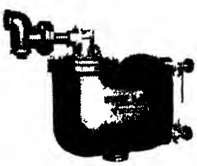


Barnes & Jones Modulation Valve. Large unobstructed passages prevent trouble from scale and dirt. Tail piece extra heavy to prevent breakage, extra long to facilitate connection to radiators. Valve is packless with quick opening, non-rising stem and renewable disc seat.

Size	½″	¾″	1″	1¼″
Cap. Sq. Ft. Rad.	30	60	100	180

Condensators

For returning water of condensation to boiler from open return line systems independently of boiler pressure, without change in operating conditions, air binding, or admitting steam to return side. Simple in construction, but positive in operation. Its few moving parts are wholly enclosed. Connections are large to eliminate friction and insure an easy and sensitive working apparatus under all conditions. All working parts are of best bronze metal.



No. 32 Condensator

No.	Capacity in Sq. Ft.
31	700
32	1,600
33	3,500
34	6,000
35	10,000
36	16,000
37	32,000

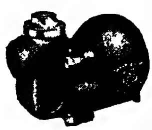
Vent Traps

Equipped with float valve to prevent discharge of water, and ball check valve to allow free discharge of air, but prevent return into system, enabling B. & J. Vapor Systems to operate under vacuum conditions, saving fuel.

Vent Trap

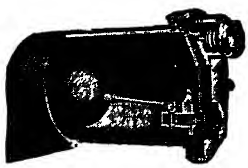
New B & J Thermostatic Trap. The interchangable control unit contains the thermostatic element, carries its own seat of special alloy and is a complete operating unit in itself. Calibrated under actual working pressure at the factory and locked in adjustment. Unit easily and quickly replaced without special tools; lift out the old unit and drop a new one in.

Size Valve	½″	¾″	1″
Pounds Water per Hour	30	80	200
Sq. Ft. C. I. Rad.	120	320	800
Sq. Ft. Coils	100	240	600



Blast Drip Traps, Type BD—For use on Unit Heaters, also drips from supply mains and risers and on returns from water heaters and indirect stacks. Float-controlled valve governs discharge of water; thermostatically-controlled valve allows passage of all air but prevents passage of steam. Made with 1¼ in. tappings. Capacity 500 lbs. per hour at ½ lb. pressure.

Blast Traps



Combination float and thermostatic traps with an air and water capacity large enough to take care of the condensation from the largest vento stacks, dry kiln coils, hot water heaters and other units condensing large quantities of steam at low pressures. Made in sizes from 1 to 2½ in. Capacities to 5,000 lbs. of water per hour. 596

Armstrong Machine Works

851 Maple Street **Three Rivers, Mich.**

District Sales Offices in 42 Cities

Exclusive Manufacturers of Armstrong Inverted-Bucket Steam Traps

A Specialized Product—For more than twenty years, the ARMSTRONG MACHINE WORKS has concentrated on the design, manufacture and application of inverted bucket steam traps. This specialization has resulted in a trap that will adequately meet any operating requirement and a sales and service organization competent to handle any steam trap problem.

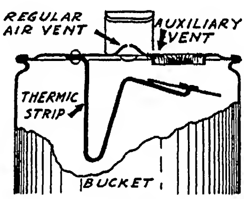
Simplicity—The Armstrong Steam Trap has only two moving parts—the valve lever assembly and the inverted bucket. Friction is practically eliminated in this mechanism. All wearing parts are made from nickel chrome steel except the discharge valve and seat which are made from a special chrome steel heat treated after machining to obtain maximum hardness and toughness. Many years of service without maintenance expense is the customary experience of Armstrong trap users.

Large Capacity—Discharge orifices used in Armstrong traps are very large in proportion to the size of the pipe connections.

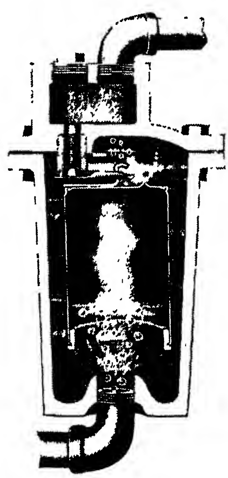
Armstrong trap capacity ratings are not theoretical but show actual test capacities when handling condensate at steam temperature. The effect of flash steam and pipe friction to and from the trap is thus automatically taken into consideration. (See chart).

Avoid Steam Trap Troubles—The customary troubles with steam traps are leaky valves, air-binding and plugging up with dirt or oil. The intermittent action of this trap and the metal used in the valves stop scoring and wire-drawing, the common sources of leakage. Air-binding is impossible because the air passes out of the bucket ahead of the steam through the vent at the top. When the trap is discharging, the flow of water under the bottom of the bucket prevents the accumulation of any dirt or sediment.

The "Blast" Type Trap—For fast handling of large quantities of air, the standard Armstrong trap can easily be furnished as a "blast" type trap by the use of a thermic bucket. The air handling capacity of this bucket is approximately 100 times as great as with the regular air vent. As long as the trap is cold a large vent in the top of this bucket remains open, and it is impossible for the bucket to float and close the valve with this vent open. As soon as the air has been eliminated and steam comes to the trap, the heat bends the strip of thermic metal supporting a flat disc which closes the large vent. Thereafter, the trap functions in the normal manner. This thermic bucket can be supplied in any size Armstrong trap. The table on the following page gives the prices for traps so equipped.



Thermic Bucket in "Blast" Trap


Cross Section of Trap Showing Regular Air Vent in Top of Bucket

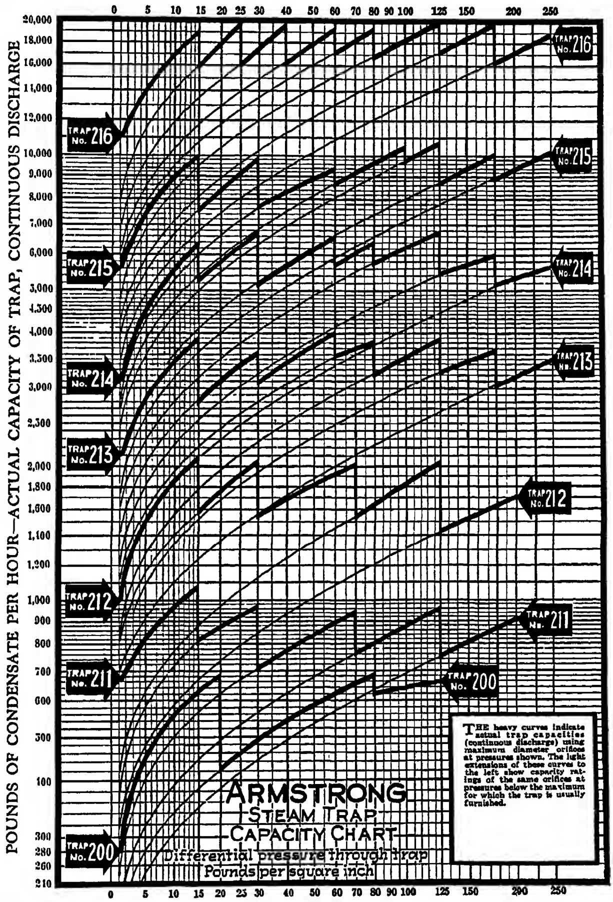
Service Organization—The satisfactory operation of all Armstrong traps is assured by 42 district representatives in the United States and Canada. Stocks of these traps are carried in nearly 100 leading cities. We will gladly put you in touch with your nearest representative.

The Armstrong Trap Book gives very complete information on trap selection, installation and maintenance. Write for your copy.

Sizes, Capacities and List Prices of Armstrong Traps

Trap Size	No. 200 and 201	No. 211	No. 212	No. 213	No. 214	No. 215	No. 216
Pipe Connections	½″	½″	½″	½″ or ¾″	1″	1″ or 1¼″	1½″ or 2″
List Price (Regular)	$7.00	$9.25	$15.00	$20.75	$29.00	$38.00	$55.00
List Price (Blast Trap)	$8.50	$10.75	$17.00	$22.75	$31.50	$40.50	$60.00
Telegraph Code (Regular)	Acacia / Acanthus	Aspen	Birch	Walnut	Hemlock	Larch	Tamarack
Telegraph Code (Blast Trap)	Acacette / Acanette	Aspette	Birette	Walette	Hemlette	Larette	Tamrette
Height	4″	6⅜″	8″	10¼″	12½″	14″	16¾″
Diameter	4¼″	4⅛″	5″	6⅜″	7½″	8½″	10$\frac{5}{16}$″
Weight	3¼ Lb.	5½ Lb.	10½ Lb.	19 Lb.	32 Lb.	47 Lb.	76 Lb.
Maximum Pressure	125	200	200	250	250	250	250

Trap capacities in the accompanying chart are *Actual Continuous Discharge* capacities. To be efficient, the trap must remove the water as it forms, which may be at a much higher rate than the average. Therefore it is good practice to allow a factor of safety of at least two in all steam trap applications. Four conditions may tend to increase the necessary factor of safety for any particular installation, (1) an excessive amount of air, (2) wide variations in rate of condensation, (3) variable steam supply pressure, and (4) variable back pressure in the return line. Many common combinations of these conditions call for the use of a factor of safety as large as six or eight.



Below—Rate of Condensation in Bare Steam Pipe at 75°F. Room Temperature.

Gauge Pressure Pounds	Temperature Difference Degrees F.	Pounds of Water per Hour per Linear Foot Pipe Sizes in Inches										Flat Surface or Larger Pipe Per Sq. Foot
		1	1½	2	3	4	5	6	8	10	12	
5	153	.16	.21	.25	0.30	0.45	0.54	0.64	0.82	1.01	1.19	0.34
50	223	.28	.38	.45	0.63	0.79	0.97	1.14	1.47	1.80	2.13	0.62
100	263	.36	.41	.59	0.84	1.05	1.28	1.51	1.94	2.39	2.82	0.82
150	291	.43	.59	.71	1.00	1.26	1.54	1.82	2.35	2.89	3.40	0.99
200	313	.50	.68	.81	1.16	1.45	1.74	2.09	2.69	3.30	3.91	1.14

C. A. Dunham Company

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Factories: Marshalltown, Iowa; Michigan City, Ind.; Toronto, Canada

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C. A. Dunham Co., Ltd.,
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Toronto, Ont., Canada

C. A. Dunham Co., Ltd.,
(of the United Kingdom)
18 St. Thomas St., S.E., London

Over eighty sales offices in the United States, Canada and the United Kingdom bring Dunham Heating Service as close to you as your telephone. These representatives are available for engineering counsel in correct selection of Dunham Systems and Appliances for any type of building. The accumulated experience of the entire Dunham organization is put at the disposal of the Heating and Ventilating Engineer. This cooperation is available for *Modernization Work*. as well as for new construction in industrial, commercial and other projects.

DUNHAM DIFFERENTIAL VACUUM HEATING

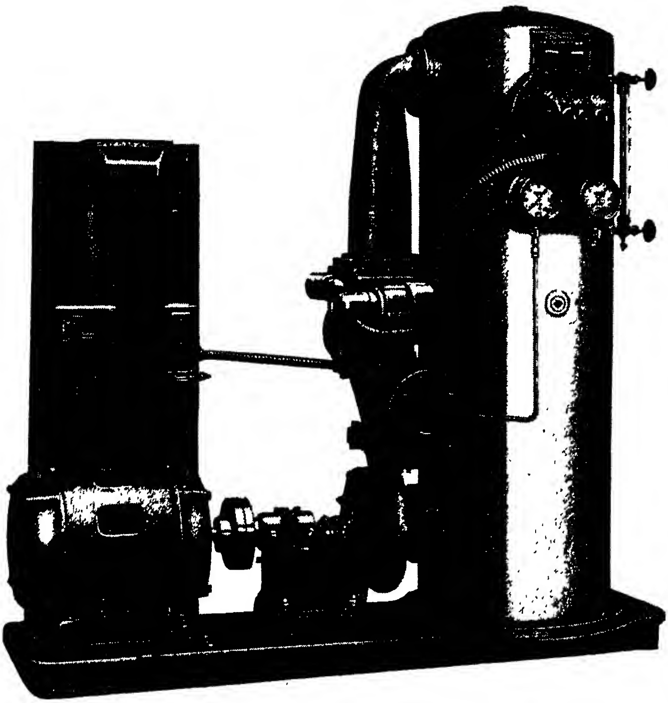
The advantages of the Sub-Atmospheric principle which govern steam distribution in Dunham Differential Vacuum Heating Systems are utilized in three different applications:

1. **The Dunham Differential Vacuum Heating System for High Duty**—This is a two-pipe system giving excellent room temperature control and air conditions. The pressure, the temperature and the volume of steam in circulation are varied under a control which is a normal function of system operation. The wide range over which the vacuum and quantity of steam *in* the radiators is regulated (from atmosphere to twenty-five inches) establishes correct rates of heat emission with radiators either complete or partially filled as required. A continuous valuation of heat requirements under positive temperature control may be secured through nine thermostats in a building, or zone of a building.

2. **The Dunham Differential Vacuum Heating System for Low Duty**—The design of this system embraces equipment of the same general type as used in the High Duty System. Its fuel economy closely approaches that system but it does not claim the same preciseness of temperature control as characterizes the High Duty System. However, the principal of control is the same as in that system; the radiator temperatures are varied and radiators may be either completely or partially filled; the vacuum, however, is limited to fifteen inches. The Low Duty System can be very

effectively related to existing buildings in changing over ordinary vacuum return line systems to differential operation. One or more thermostats may be used with a Low Duty System.

3. **The Dunham One-Pipe Vacuum Heating System with Sub-Atmospheric Steam** in which the range of steam temperatures and pressures is ample to give great flexibility of operation in the lower range of vacuums. This System is designed primarily to enable owners of existing one-pipe systems to obtain the benefits of the Dunham Differential Vacuum principle of operation by rearranging the system to operate on that principle. The change-over can be made without extensive cutting of floors or walls. Systems having air lines will usually require no such cutting.



"DV" Series for Dunham Differential Vacuum Heating Systems
"VR" Series for Vacuum Return Line Heating Service

THIS line of pumps is characterized by an entirely new and improved Vacuum Producing Element of exceptional quietness and high efficiency; by a Discharge Valve which is rugged and reliable and by a new Control Equipment Mounting. These, with other refinements, result in an outstanding piece of equipment.

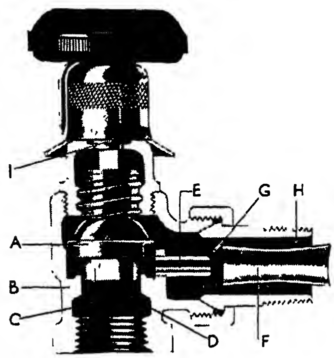
These Pumps are built in eleven sizes, ranging from capacity of 2,500 to 150,000 sq. ft. of radiation inclusive. They meet the requirements of the Vacuum Return Line Heating Pump Manufacturer's Section of the Hydraulic Institute.

D. G. C. Trap and Valve Company, Inc.

9 East 46 Street . New York Plaza 3-3790

Makers of Cryer Radiator Control Valves, Radiator Traps and Steam Traps
Complete Vapor Heating Systems

Distributors in All Principal Cities

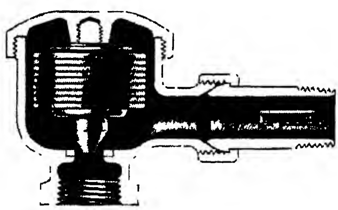


Cryer Radiator Control Valve

A Control Valve which gives to steam heating systems the same operating economies, and the same medium temperatures in the radiator afforded by hot water heating systems.

The Cryer Control Valve mixes the steam with air already in the radiator, and circulates this humid mixture of air and steam through all sections of the radiator.

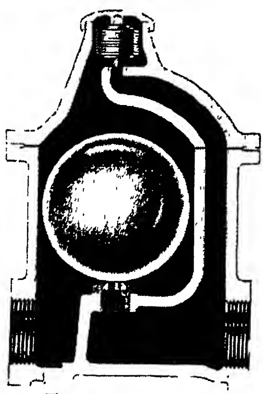
The temperature of the whole radiator may be varied at will from the low temperatures usual only with hot water, to the full steam effect.



Cryer Radiator Trap

A Thermostatic Radiator Trap constructed to endure as well as to operate efficiently. The bellows is made of a special bronze alloy, the seat of stainless steel, the spud, nut and cap are bronze forgings, and the body is a heavy bronze casting. Made to standard dimensions.

Suitable for low pressure, vacuum or vapor heating.



Cryer Condensation Trap

A heavy duty Condensation trap for Unit Heaters, Ventilating Stacks, Hot Water Heaters and other large units. Compact design, small enough to be used anywhere, but of large capacity. Cannot air-bind.

The Cryer Condensation Trap is of the unattached ball float type, with a thermostatic air by-pass. The valve seats are stainless steel, the float seamless copper nickel-plated, the bellows of bronze alloy, and the body of heavy cast-iron.

The pressure limit is 10 lbs.

Catalogues on Request

GRINNELL COMPANY INC.

Heating, Industrial and Power Plant Piping, Fittings, Hangers, Valves, Pipe Bending, Welding, Piping Supplies, Etc.

Executive Offices: **Providence, R. I.**

National Distributors of Thermoflex Traps and Heating Specialties

For data on other Grinnell Products, see pages 700-703

Thermoflex Specialties

The heart of all Thermoflex Traps is the Hydron Bellows.

The Hydron Bellows is formed under hydraulic pressure. This powerful internal pressure locates any weakness of any nature in the tubing. Such hydraulic pressure is many times more severe than any pressure the Trap will ever be called upon to control. Every Thermoflex Trap, therefore, is practically indestructible.

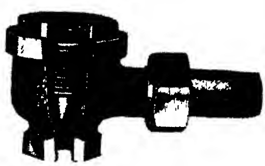
Thermoflex Traps have an exceptionally large orifice. This large orifice combined with high lift, insures fast action and freedom from clogging.

We supply Thermoflex Traps guaranteed for steam pressures up to 25 lb. and to 125 lb. Complete information and details of typical installations will be gladly sent on your request. Ask for Data Book on Thermoflex Heating Specialties.

Valves, Traps, Gauges, Etc.

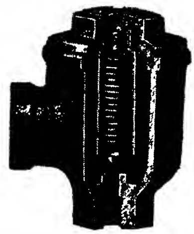
The Thermoflex line includes: Radiator Traps, Offset Traps, Blast Traps, Drip Traps, High Pressure Traps, Vent Traps, High-grade Packless Inlet Valves, and the Thermoflex Alternator, Thermoflex Compound Gauge, Thermoflex Damper Regulator.

No. 12 Thermoflex Radiator Trap



The full eight-fold Thermoflex-Hydron Bellows is the best bellows ever made. Because of the Hydron-forming process every bellows is absolutely perfect. Body is heavy bronze construction throughout. Fully nickel-plated with highly polished trimmings. The No. 12 is made in angle and in corner patterns, with ½ in. inlet and ½ in. outlet tappings. The inlet neck is double thick to allow for expansion strains. Guaranteed for steam pressures up to 25 lb.

Thermoflex High Pressure Traps



The No. 100 Thermoflex Trap is guaranteed for steam pressures from 25-125 lb. Must not be used where the steam temperature exceeds 400 deg. Fahr.

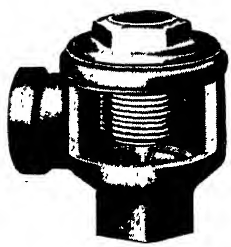
For use with all types of process work, Laundry Machinery, Kitchen Equipment, Hospital Sterilizers, Vulcanizers, Dry Kilns, Unit Heaters, Street Steam Service, etc., in fact any place that a trap is desired for service at the above pressures.

Small, compact and inexpensive in comparison to the usual float or bucket trap.

Extra heavy body. Renewable nickel steel seat and disc. Bellows made from special bronze tubing and encased in brass sleeve to prevent distortion due to pressure.

Regularly furnished without unions, plain nickel finish. Can be furnished with unions, polished nickel or chromium plated at extra cost.

No. 4 Thermoflex Drip Traps



For dripping mains, risers, coils and unit heaters, we offer this type of trap. Cast iron body, bronze cap and inserted renewable bronze seat, angle pattern only, without unions. Can be used for any general purpose where a finished, nickel-plated trap is not necessary, and at a lower cost. Guaranteed for steam pressures up to 25 lb.

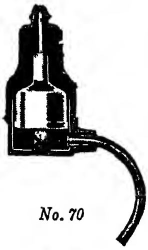
Hoffman Specialty Co., Inc.

Main Office and Factory, **Waterbury, Conn.**

General Sales Department, Chrysler Building, **New York**

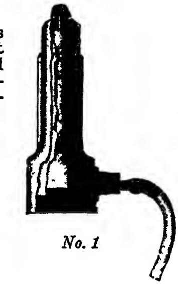
Sales Representatives in Principal Cities

VENTING VALVES FOR ONE-PIPE STEAM SYSTEMS



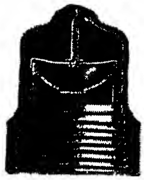
No. 70

No. 70—Airport—Is an efficient, low cost air valve—made and guaranteed by Hoffman. Radiator connection 1/8 in.



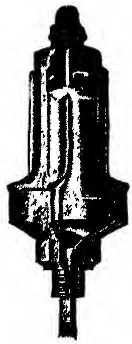
No. 1

No. 1—Hoffman Siphon Air Valve—The "perfect" venting valve, with patented double-shell construction. Angle pattern only. Radiator connection 1/8 in.



No. 44

No. 44—Quick Vent Valve—For venting mains, risers, coils, etc., where water is not encountered. Vent port—1/16 in. Connection 1 in. female. Where water is liable to be a factor use No. 5 Float Vent. Connection 3/8 in. Vent port 1/16 in. or on order, 3/16 in. for pressures below 3 lbs.



No. 5

No. 71

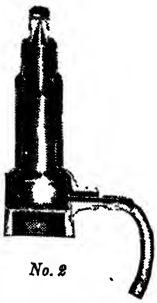
No. 71—Airport—Same as the No. 70, except with straight shank 1/8 in. connection for venting certain types of concealed radiators.

Maximum Guaranteed Operating Pressure—10 Lbs.

VENTING VALVES FOR ONE-PIPE VACUUM SYSTEMS

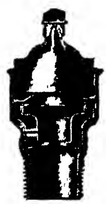
No. 77

No. 77—Vacuum Air Valve—With double air locks, consisting of air check and vacuum diaphragm, selling in the lower price field. Angle pattern, 1/8 in. connection.



No. 2

No. 2—Hoffman Siphon Air and Vacuum Valve—Air is freely vented without steam or water loss. The double "air lock"—air check and powerful vacuum diaphragm—prevents return of air. Angle pattern only. Radiator connection 1/8 in.



No. 16

No. 16—Vacuum Vent—For venting risers, coils or mains, that end 18 in. or more above boiler water line. Vent port, 1/16 in. Connection 3/4 in. Where mains end less than 18 in. above water line, use No. 6 Vacuum Float Vent. No. 6 connection 3/8 in. Vent port 1/16 in. (standard) or 3/16 in. on order for pressures below 3 lbs.

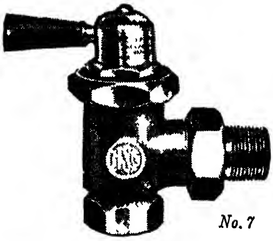
No. 78

No. 78—Is the same as the No. 77 except with 1/8 in. straight shank connection for venting certain types of concealed radiators.

No. 6

Maximum Guaranteed Operating Pressure—10 Lbs.

MODULATING AND PACKLESS SUPPLY VALVES



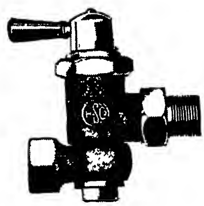
No. 7

The No. 7 Hoffman Adjustable Modulating Valve—For use in Vapor, Vapor Vacuum or Forced Hot Water Systems, is made in $\frac{3}{4}$ in. size, angle or swivel pattern, having a range of adjustment up to 200 sq. ft. of direct cast-iron radiation.

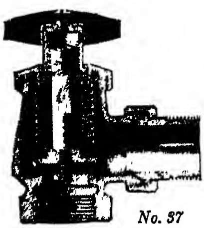
After installation, whether the system is in operation or cold, the port of each valve is adjusted for size of the radiator to which it is attached. Adjustment is simple; loosen a locknut; turn valve handle until proper graduation is opposite notch on bonnet; then tighten locknut. The valve handle may then be moved to admit sufficient steam to heat a quarter, half, three-quarter, or entire radiator. The valve steam stuffing box has a frictionless metallic fibre packing that will last indefinitely and require no attention, giving at the same time, a valve action so free that the pressure of only one finger is required to open the valve.

The No. 7 Hoffman Modulating Valve provides a most convenient and accurate method of obtaining balanced distribution throughout vapor or vacuum systems and control of the output of each heating unit.

The visible adjustment aids the designing engineer by enabling heating contractor to make a final accurate adjustment which compensates for slight irregularities in pipe sizes, failure to ream pipe, installation of extra fittings not foreseen in original layout, etc. The advantages of an adjustable port in forced hot water systems to secure proper balance makes the **No. 7 Valve** especially adaptable for such use.



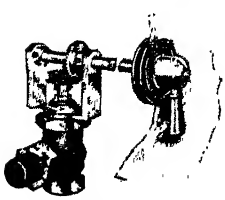
No. 7 Swivel Connection



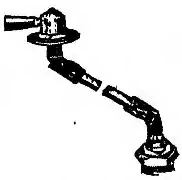
No. 37

No. 37—Bellows Packless Valves are used on one- or two-pipe systems. The durable, leak-proof bellows is hydraulically formed and eliminates steam or water leakage and leakage of air into vacuum systems. There is no torsional strain on bellows, as flat-sided stem operates thru key in bonnet. An attractive hexagonal Bakelite handle is standard on all sizes, or the $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. sizes can be equipped with lever handles at no extra charge. Angle pattern (standard) all sizes—$\frac{1}{2}$ in. to $1\frac{1}{2}$ in. Swivel connections for $\frac{1}{2}$ and $\frac{3}{4}$ in. valves at slight extra cost. Adjustable orifice plates available for $\frac{3}{4}$ in. size only.

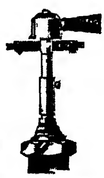
SPECIAL ATTACHMENTS FOR NOS. 7 AND 37 VALVES



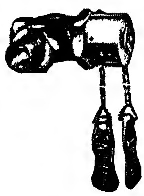
Bevel Gear for Nos. 7 and 37



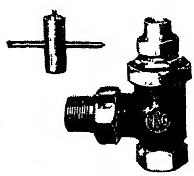
Offset Extended Nos 7 and 37



Straight Stems Nos. 7 and 37

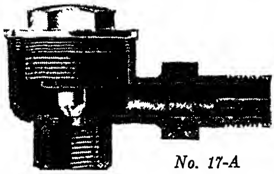


Chain Pull for No. 7 only



Lock Shield for Nos. 7 and 37

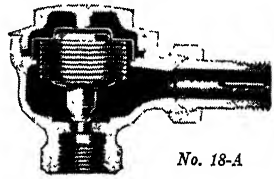
BELLOWS TYPE THERMOSTATIC TRAPS



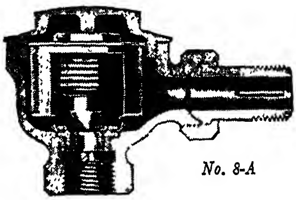
No. 17-A

The No. 17-A Radiator Trap has a hydraulically formed bellows thermostat attached to cap. Hydraulic forming process automatically places severe pressure test on bellows. Valve pin is of special nickel-silver alloy. Capacity, 200 sq. ft. direct radiation. Valve port, $\frac{3}{8}$ in. Maximum guaranteed operating pressure, 15 lbs. Angle pattern or with swivel connection. Connections $\frac{1}{2}$ in.

The No. 18-A Radiator Trap is also equipped with the "tested-in-the-making" hydraulically formed bellows. Thermal members are interchangeable in all No. 18-A bodies without adjustment. Valve port, $\frac{3}{8}$ in. Nominal capacity at pressures below 1 lb.—300 sq. ft. direct radiation. Angle pattern with $\frac{1}{2}$ in. connections standard. Also furnished with $\frac{3}{4}$ in. outlet and with swivel connection. Maximum guaranteed operating pressure, 25 lbs.

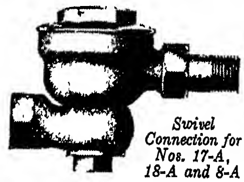


No. 18-A



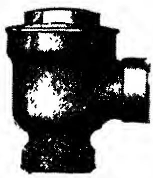
No. 8-A

A novel feature of the **No. 8-A Radiator Trap** is the combination of all vital parts—bellows, valve pin and valve seat, into a single unit which threads into the body and the joint is sealed by a soft copper gasket. The valve pin and renewable seat is of a special wear-resisting nickel-silver alloy. Valve port, $\frac{3}{8}$ in. Nominal capacity at pressures below 1 lb.—300 sq. ft. direct radiation. Angle pattern with $\frac{1}{2}$ in. connections standard. Also furnished with $\frac{3}{4}$ in. outlet and with swivel connection. Maximum guaranteed operating pressure, 25 lbs. or 50 lbs. on special order.



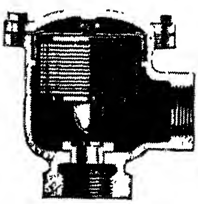
Swivel Connection for Nos. 17-A, 18-A and 8-A

The swivel connection provides straightway, right or left hand offset or any intermediate angle. The fitting is reversible, giving the option of two roughing-in dimensions.



No. 9-A

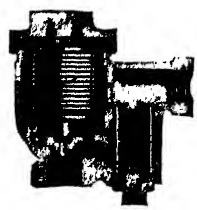
No. 9-A is similar to the No. 8-A in construction, except with $\frac{3}{4}$ in. inlet and outlet connections. Furnished in angle pattern only, with or without union connection. Valve port, $\frac{7}{16}$ in. Nominal capacity at pressures below 1 lb.—700 sq. ft. direct radiation. Maximum guaranteed operating pressure, 25 lbs. or 50 lbs. on special order.



No. 10-A

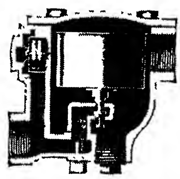
The No. 10-A is a heavy duty thermostatic trap for dripping unit heaters, short steam mains, risers, etc., where a cooling leg (for temperature drop) can be installed. The bellows thermal member is water-filled to enable it to withstand severe service, and is interchangeable in No. 10-A bodies without adjustment. Nominal capacity, 2800 sq. ft. direct radiation. Valve port, $\frac{9}{16}$ in. Iron body—angle pattern—without union. Maximum guaranteed operating pressure, 25 lbs. Connections 1 in.

The Nos. 20 and 21 Thermostatic Steam Traps operate on pressures from 0 to 100 lbs. without change or adjustment. Bodies are all bronze with integral strainer. The hydraulically formed bellows is water-filled to better withstand the shocks usually encountered in high pressure service. Thermal members interchangeable without adjustment. Valve pin and renewable seat is of special nickel-silver alloy. No. 20—$\frac{1}{2}$ in. connections—$\frac{3}{16}$ in. valve port. No. 21—$\frac{3}{4}$ in. connections—$\frac{1}{4}$ in valve port.



Nos. 20 and 21

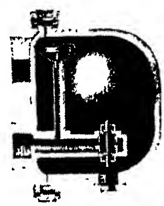
FLOAT AND THERMOSTATIC TRAPS



No. 24

The No. 24 Drip Trap (float and thermostatic) is used for draining risers, short steam mains, unit heaters, etc. Weighing only 4¾ lbs., it has a capacity of 780 lbs. of condensate per hour at 1 lb. pressure difference. The combination of large capacity, light weight and compactness is made possible by its balanced, double-seated valve assembly, which permits the use of a direct acting float. Iron body, with ¾ in. or 1¼ in. connections. Maximum guaranteed operating pressure, 15 lbs.

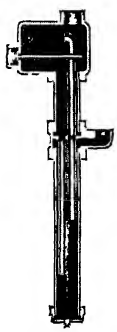
The principle of the balanced, double-seated valve assembly, with direct acting float, is also used in the Heavy Duty Float and Thermostatic Traps Nos. 25 to 28. The smaller trap has a capacity of 2250 lbs. of condensate per hour at 1 lb. pressure difference and is furnished with 1 in. connections (No. 25) or 1¼ in. connections (No. 26). The No. 27—1½ in. and 28—2 in. traps have a capacity of 3090 lbs. of condensate at 1 lb. pressure difference. These traps are used for draining mains, hot-water generators and other large capacity units where the operating steam pressure does not exceed 30 lbs.



Nos. 25, 26, 27 and 28

HOFFMAN "CONTROLLED HEAT" EQUIPMENT

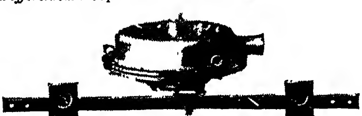
WITH DIFFERENTIAL LOOP



Differential Loop

Hoffman "Controlled Heat" is a two-pipe vapor-vacuum system of heating, using the No. 7 Adjustable Orifice Modulating Valves for balancing the distribution of steam and controlling the radiator output. Hoffman traps are used at the return end of the radiators. Where hard coal is used as fuel and the installation does not require a normal operating pressure in excess of eight ounces, the Differential Loop equipment is recommended. The Loop is a non-mechanical device which insures the return of condensate to the boiler should the pressure exceed that at which it will return by gravity.

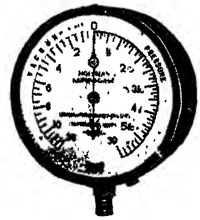
No. 15



Damper Regulator

The No. 15 Valve is used only in conjunction with the Differential Loop and functions to relieve the air from the entire system and prevent its return.

The No. 13 Damper Regulator is extremely sensitive in controlling the drafts to meet slight changes in the demand for steam. A compensating plate prevents the accumulation of water on the diaphragm when it is depressed, maintaining perfect balance at all times. Connection, 1 in.



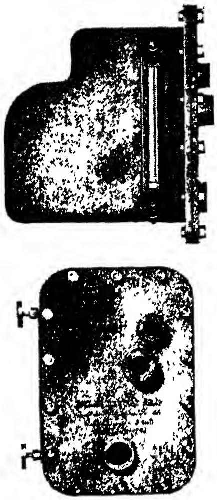
Kompo Gage

The No. 14-A Kompo Gage measures pressure up to 30 lbs., vacuum to 30 inches. The pressure side is graduated in ounces up to 5 lbs. and vacuum is shown in half-inch graduations up to 10 inches. Connection ¼ in.

HOFFMAN "CONTROLLED HEAT" EQUIPMENT

WITH BOILER RETURN TRAP

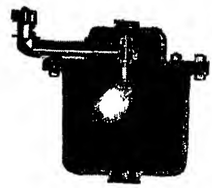
For Hoffman "Controlled Heat" installations fired with oil or gas burners, or soft coal, where pressures are generally built up quickly, or where Unit Heaters or other devices requiring a constant pressure in excess of 8 ounces, are used, we recommend the Hoffman Boiler Return Trap equipment. At low pressures, water returns by gravity to the boiler but when the pressure exceeds this point, the Boiler Return Trap functions to maintain the prompt return of condensate to the Boiler. The balanced, double-seated valve mechanism opens quickly when the high water level of the trap is reached, allowing rapid equalizing of pressures.



Boiler Return Trap

Air is freely vented thru the Receiver Vent but is not permitted to return to the system thru the valve.

Boiler Return Traps and Receiver Vents are made in four sizes of the following capacities: No. 30—2000 sq. ft. No. 31—2900 sq. ft. No. 32—5300 sq. ft. No. 33—8000 sq. ft. These capacities are based on a condensation rate of ¼ lb. per sq. ft. of radiation per hour and one operation of the trap per minute.



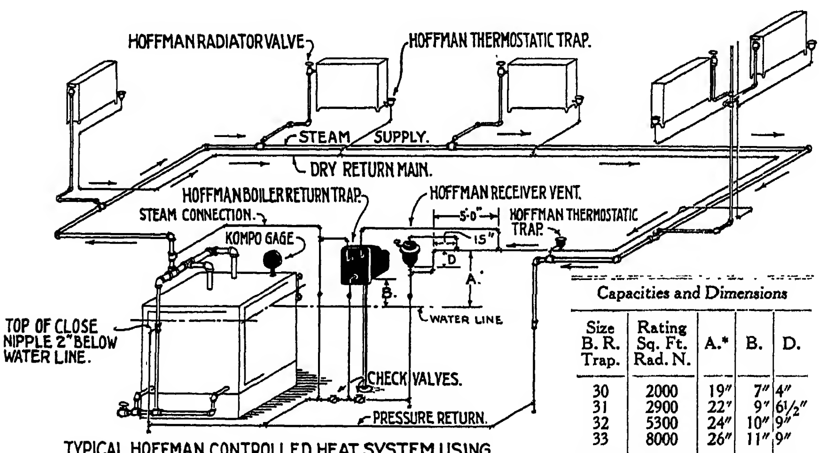
Receiver Vent

The sketch below shows the general piping arrangement where the Boiler Return Trap and Receiver Vent are installed.

The Receiver Vent provides ample storage capacity for the condensate which accumulates during the period that the Trap is discharging to boiler and prevents temporary flooding of the dry returns. Minimum differential between the boiler water line and low end of dry return is therefore required.

Operation of the Return Trap is noiseless as all working parts are enclosed and the balanced valve construction permits the use of light counter-weights. Interior parts are made of non-corrosive materials. Detailed literature supplied on application.

TYPICAL LAYOUT USING BOILER RETURN TRAP EQUIPMENT



TYPICAL HOFFMAN CONTROLLED HEAT SYSTEM USING BOILER RETURN TRAP AND HAVING PRESSURE RETURN.

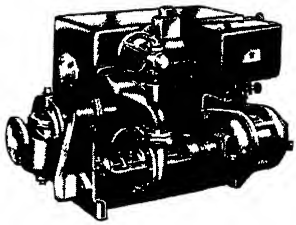
Capacities and Dimensions

Size B. R. Trap.	Rating Sq. Ft. Rad. N.	A.*	B.	D.
30	2000	19"	7"	4"
31	2900	22"	9"	6½"
32	5300	24"	10"	9"
33	8000	26"	11"	9"

*Minimum to Bottom of Pipe.

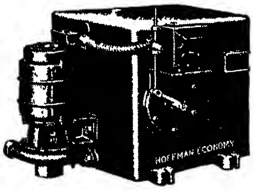
HOFFMAN-ECONOMY PUMPS

Vacuum Pumps—The Jet-Type Vacuum Producer enables this pump to deliver full rated capacity in extremely hot-water. There are no moving parts with close clearances. Single pump consists of one vacuum producer, centrifugal pump, cast-iron receiver and accumulator tank, automatic boiler feed control, motor and full automatic starting equipment, mounted complete. Duplex units consist of a single cast-iron tank, with two vacuum producers, two pumps, two motors, each pump and motor furnished with full automatic control. Standard pumps made in pressure of discharge range of 20—30 and 40 pounds. Capacities 2500 to 300,000 sq. ft. cast-iron radiation.



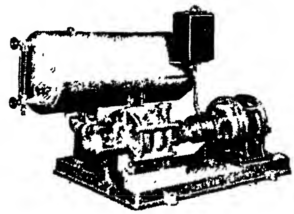
Single Vacuum Pump

Type E Condensation Pump is a compact unit shipped ready for installation. It consists of pump, motor, cast-iron tank with all necessary tank trimmings for automatic control. Pump is of close coupled type with impeller securely mounted on the motor shaft extension. These pumps give full capacity at 1750 r.p.m. and improved performance makes possible the use of comparatively small motors with consequent low consumption of electricity. Capacities, 1,000 to 20,000 sq. ft. cast-iron radiation. Discharge pressure for standard pumps 10—20—30 pounds. Discharge pressure up to 60 pounds can be furnished.



Type E Pump

Type B & C Condensation Units consist of pump, motor, steel tank and tank trimmings assembled on a single cast-iron base. The pump is direct connected to the motor through a flexible coupling. The pumping unit is of the multi-stage type and so constructed that access may be had to impeller and other interior parts without breaking the piping connections. Receiver is of heavy gauge welded steel securely attached to cast-iron feet by anchor bolts welded to the shell. Capacities range from 6,000 to 70,000 sq. ft. of cast-iron radiation, with discharge pressure up to 175 lbs. duplex or single units can be furnished.



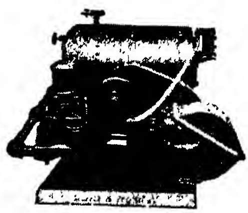
Type B and C Pumps

Vertical Underground Pumps and Receivers are designed for use on heating plants where the returns are located below basement floor levels, or otherwise too low for horizontal condensation pumps. The receiver is of heavy cast-iron suitable for underground use without danger of corrosion, making a cement lined pit unnecessary.

The pumps are of vertical bronze fitted centrifugal type suspended from the receiver cover and bearings may be renewed without removing impeller from shaft or shaft from pump. A flexible coupling connects motor to pump shaft. Shaft is protected from bearing wear by a removable sleeve. Receiver is equipped with float switch for automatic control of motor, this control being removable without disturbing pump or pipe connections. Duplex Units with two pumps and motors mounted in a single receiver of larger size can be furnished if desired. Capacities, 3000 to 30,000 sq. ft. of cast-iron radiation with discharge pressures ranging from 10 to 100 lbs.

Vertical Type

The Reciprocating Pump and Receiver is recommended where high pressures of from 50 to 100 lbs. are met and moderately priced equipment is desired. These pumps operate on less power than required for centrifugal pumps, but are not as quiet, due to a slight pulsation at the end of each stroke and slight hum of the chain drive. As the pumps are usually installed in Laundries, Drying Plants or other industrial establishments, the slight noise is not objectionable. Receivers are of welded steel equipped with automatic float control. Capacities 1,000 to 10,000 sq. ft. Discharge pressures 50 to 150 pounds.



Reciprocating Pumps

William S. Haines & Company

12th and Buttonwood Sts., **Philadelphia, Pa.**

Manufacturers of Equipment for Vapor and Vacuum Heating Systems

Haines Radiator Traps.

Haines Medium Pressure Thermostatic Traps.

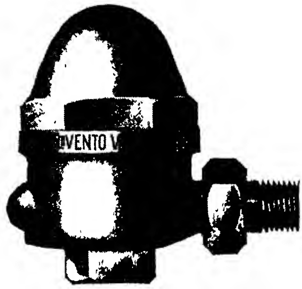
Haines High Pressure Thermostatic Traps.

Haines Float and Thermostatic Blast Traps.

Haines Vent Traps.

Haines Modulating Valves.

Haines Boiler Return Trap.



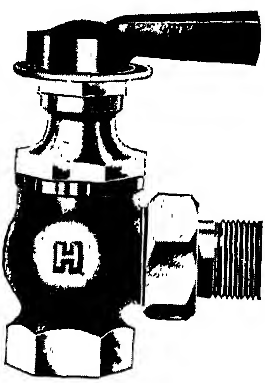
All HAINES traps, whether designed for pressures below atmosphere or pressures in excess of 100 lbs. per sq. in., employ as their operating member a specially constructed bourdon tube—the principle that actuates the steam gauge.

The tube is of tempered steel. It is charged with a volatile fluid and hermetically sealed. It is the expansion and contraction of the fluid, under varying temperatures, that furnishes the operating power.

The tube is mounted vertically on a horizontal valve motion. The end opposite the valve is anchored so that the travel of the tube either opens or closes the valve piece.

The thermostatic member is outboard the valve seat and closes the valve against the flow of steam. This arrangement prevents fouling of the trap due to scale or other foreign matter and permits a thorough draining of the unit to which it is attached.

HAINES thermostatic traps are made in sizes from ½ in. to 1½ in. All traps are factory tested and adjusted before shipment.



HAINES modulating valves never need repacking. They seat tightly and open on less than a full turn of the lever or wheel handle.

Made in sizes from ½ to 2 in. in angle, globe, or corner pattern.

HAINES VENTO TRAPS

No. of Trap	Center to Inlet	Center to Outlet	Capacity Sq. Ft.
1	2⅞"	1 1/16"	125
2	3¼"	1 5/16"	200
2E	3⅜"	1 5/16"	250
3	3⅝"	1 11/16"	400
3E	3⅞"	1 11/16"	500

HAINES MODULATING VALVES

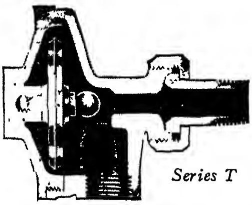
Size of Valve	Center to Inlet	Center to Outlet
½"	1 7/16"	2½"
¾"	1½"	2⅝"
1"	1 13/32"	3"
1¼"	1 15/16"	3⅝"
1½"	2 1/16"	4¼"
2"	2⅝"	4 15/16"

Illinois Engineering Company

General Offices and Factory: **Chicago**

Branches and Representatives in Principal Cities

Illinois Thermo Radiator Trap



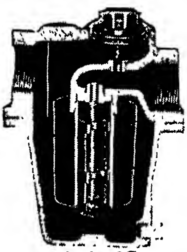
Series T

The Original Vertical Seat Trap. Self cleaning, non-adjustable. Positive and sensitive in operation. Thousands in use for over fifteen years without diaphragm replacements. Furnished in all sizes and patterns.

Illinois Modulating Supply Valve

Quick-opening, packless. Steam tight on 50 pounds pressure. Large diameter of thread spool and machine cut threads make valve operation easy. Furnished in a complete line of sizes and patterns.

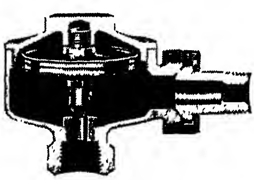
Illinois Combination Blast Traps



Series 8G

Unsurpassed for draining ventilating units, unit heaters, and for dripping mains and risers—wherever it is desirable quickly to vent air from the main as well as handle the water of condensation in quantity, whether hot or cold.

Illinois Thermo Radiator Traps

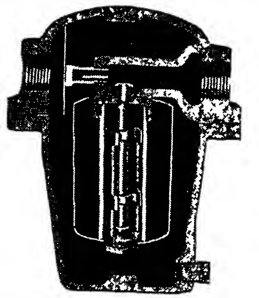


Series G

Illinois Thermo Radiator Traps for vacuum, vapor and low pressure heating systems. Has cone type valve. Flushes thoroughly and seats perfectly at all times. Valve and seat are of Nitralloy. The duplex diaphragm is of special phosphor bronze. Scientific design and rugged construction assure flexibility and long life. These diaphragms have withstood over three million strokes on a breakdown test. Will withstand 75 pounds steam pressure without damage although only low pressure service are required of them.

Made in three sizes from ½-in. to 2-in., in a variety of patterns. Furnished in either plain or nickel plated finish.

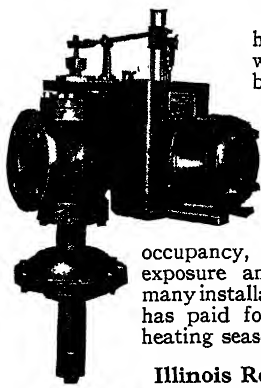
Illinois Steam Trap



Series 30

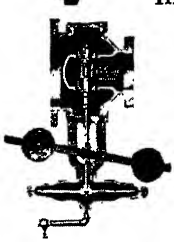
Valve and stem are separate from the bucket and operated only by the bucket at the extreme top and bottom of travel—result—*valve is always either full open or tight closed.* No wire drawing or cutting of valve and seat which are of Monel metal. Steam tight and long lasting.

Illinois Thermal-Zone Control



Prevents overheating and fuel waste in large buildings or groups of buildings heated from one central power plant. Buildings may be zoned as to occupancy, time, location, exposure and so on. In many installations this valve has paid for itself in one heating season.

Illinois Reducing Valve



In general use on vacuum or low pressure heating systems. Will reduce to 4 oz. pressure from an initial pressure of 150 lbs. The large diaphragm insures sensitive operation. Made in both straightway and expanded outlet bodies in sizes from ¾ in. to 12 in.

Kieley & Mueller, Inc.

Established 1879

Engineering Specialties for Pressure and Level Control

34 West 13th Street, **New York, N. Y.**

Factory: NEWARK, N. J.

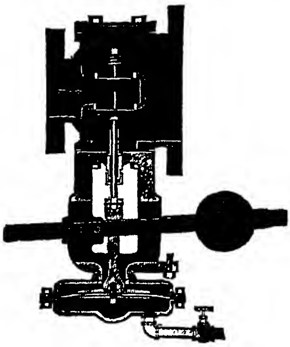
Agents in All Principal Cities

PRODUCTS—Valves: Altitude, Stop and Check, Pressure Regulating, Float, Pilot Reducing, Back Pressure, Tank Control.

Liquid Level Controllers, Pump Governors, Steam Traps, Strainers.

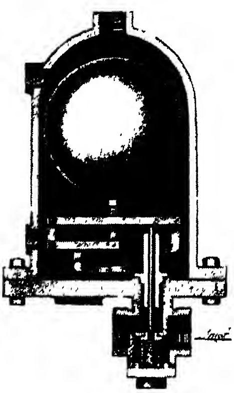
Also Damper Regulators, Hot Water Temperature Controllers, Oil Separators, Steam Separators, Return Traps, Water Columns, etc.

Catalogs sent upon request



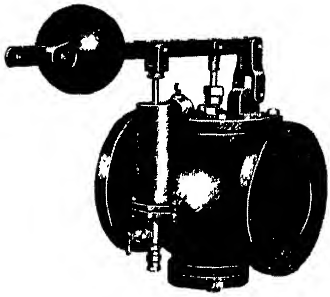
Pressure Regulating Valve

Spring and lever weighted valves for all services and for initial pressures up to 200 lbs. and reduced pressures from ½ lb. to three-quarters of the initial pressure. Single or double seated in sizes ⅜″ to 16″. Suitable for steam, water, air, oil and gas. Controlled by a small feeler pipe connected from diaphragm to low pressure side.



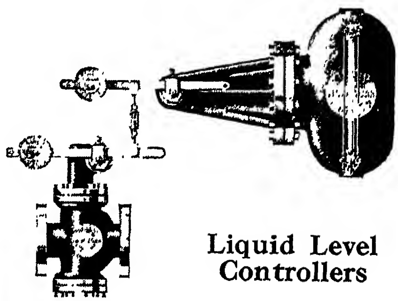
Water Feeder

Automatically maintains water level in low pressure steam boilers, receiving tanks, feed water heaters, etc. Will hold water level with little or no fluctuation. For working steam pressure up to 50 lbs.; and water pressures up to 100 lbs.



Back Pressure and Atmospheric Relief Valve

For use where plant is operated either condensing or non-condensing. Outside air dash pot insures noiseless operation. Maintains exhaust line back pressure from 0 lbs. to 25 lbs. Made horizontal or vertical lever and weight or spring operated.



Liquid Level Controllers

For the accurate control of liquids in tanks or other vessels; suitable for use in industrial plants, gasoline plants, refineries, etc. Direct connected or remote control; ball bearing spindle and easy-to-pack stuffing box; rotary or sliding valve. Write for special bulletin C-3.

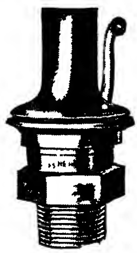
J. E. Lonergan Co.

207 Florist Street, **Philadelphia, Pa.**

Pop Safety Valves; Relief Valves; Steam Gauges; Hydraulic Gauges; Air Gauges; Water Gauges; Pressure and Temperature Gauges; Test Gauges; Gauge Boards; Oil Gauges; Clocks; Counters; Gauge Cocks; Steam Gauge Syphons; Lubricating Specialties.

Model "ORV"

Oil Relief Valve. Sizes 3/8 to 2 in.



Model "HHU"

Pop Safety Valve A.S.M.E. "House Heating Boiler."

Standard pressures, 5. 10 and 15 lb.

Model "WRV"

Water Relief Valve for tank service.

Sizes 3/8, 1/2 and 3/4 in.

Model "VAK"

Special valve for vacuum breaking.

Six sizes, 1/2 to 2 in.

Model "GLP"

Low pressure, Iron Body, Brass Mounted.

Set to blow off at 10, 15, 20, 25 or 30 lb.

Five sizes–2 1/2 to 4 1/2 in.

Model "U"

Relief Valve. Snifter, Water or Cylinder—Bronze.

Recommended for steam engines, pumps, pipe lines, etc.

Ten sizes, 1/2 to 4 in.

Model "GV"

Vacuum Gauge.

Gauges graduated to 30 in. vacuum.

Ten sizes, 2 1/2 to 10 in. dial.

Model "GOZ"

Vapor Gauge.

Graudated to 5 lb. by ounces.

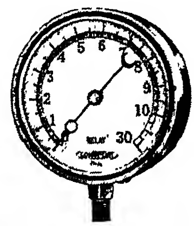
Two sizes, 4 1/2 and 5 in. dial.

Model "BLGB"

Tank in basement type of gauge for closed heating systems.

Made in two- and three-story calibrations only.

One size, 3 1/2 in. dial.



Model "BLGR"

Gauge for indicating height of water in feet.

Graduation 70 ft.

Three sizes, 3 1/2, 4 1/2 and 5 in dial.

Model "BLGW"

For either pressure or altitude, or combination of both.

Graduation 30 lb., 70 ft.

Two sizes, 3 1/2 and 4 1/2 in. dial.

Model "BLGA"

Pressed steel case gauge for indicating height of water in feet. Graduation 70 ft.

Three sizes, 3 1/2, 4 1/2 and 5 in. dial.

CATALOGUE

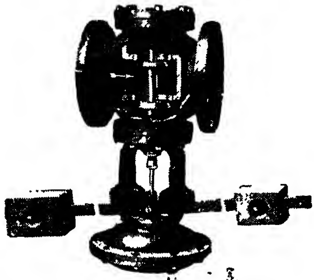
Write for our new 100-page catalogue, describing and illustrating the complete "Lonergan Line" or ask us about specialties in which you are interested.

Mueller Steam Specialty Co., Inc.

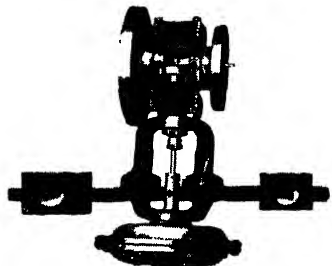
349-351 West 26th Street, New York City

Steam, Water, Air, Oil and Gas Specialties for Heating and Power Plants

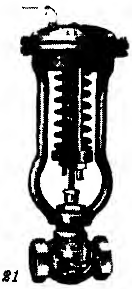
Pressure Reducing Valves—Straight Pattern and With Increased Outlet



No. 11



No. 17



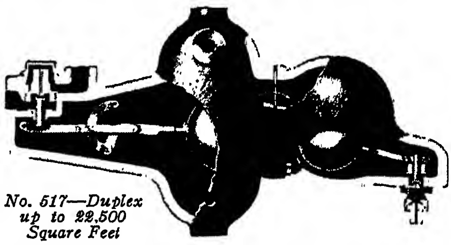
No. 21

No. 11—For Vacuum, Vapor and Low Pressure Heating Systems. Initial Pressures, up to 200 lb.; Reduced Pressures, 0 to 10 lb.

No. 17 and 21—For automatic control of reduced pressures on dead-end service, requiring a tight closing valve, such as tank heaters, kitchen utensils, sterilizing apparatus, laundry equipment, kettles, cookers, driers, etc. Initial Pressures up to 200 lb. Reduced Pressures 0 to 150 lb.

Constructed with full globe bodies. Center guide eliminates the wings on discs, and increases efficiency, assures minimum noise and prolongs the life of the seats and discs. Lever and weight operates on a steel roller bolt, assuring a most sensitive valve. Spring type furnished with special long springs for sensitive operation and wide ranges of reduced pressures.

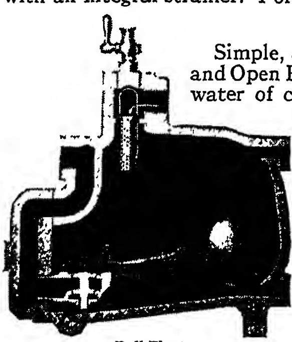
Automatic Water Feeders



No. 517—Duplex up to 22,500 Square Feet

With a powerful leverage to control the water line in steam boilers, etc. They supply make up water to compensate for evaporation, leaks, steam utilized in process work and condensation wasted. Where condensation held up in the system eventually returns in large quantities, our Duplex type protects the boiler against flooding. All working parts of non-corrosive metal, are accessible without breaking pipe connections. Provided with an integral strainer. For steam pressures up to 100 lb., water pressures up to 120 lb.

Steam Traps



Ball Float

No. 219—Up to 30 lb.
No. 221—Up to 150 lb.

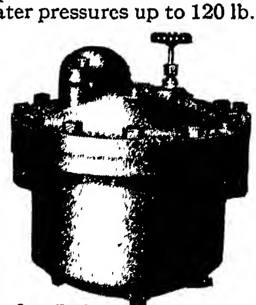
Simple, Sturdy and Compact Ball Float and Open Bucket Steam Traps for draining water of condensation from steam apparatus and steam mains.

Powerful leverage enables them to take care of large quantities of condensation.

Equipped with strainer, water gages, air cocks, blow off and integral by-pass valve, when desired.

All working parts are accessible without disturbing any pipes.

Valves are sealed with several inches of water, making the escape of steam impossible.



Open Bucket

No. 229—Up to 30 lb.
No. 231—Up to 150 lb.
No. 233—Up to 250 lb.

Sizes from ½ to 3 inches.

Catalogue and Bulletins covering our Complete Line gladly furnished on application.

New York Air Valve Corporation

476-478 Broome Street

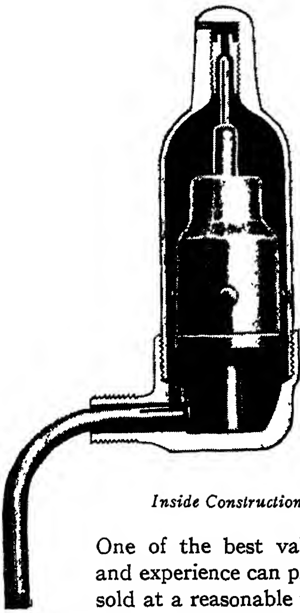
New York

THE "AIR-OUT" LINE

SYPHON **THERMOSTATIC** **NON-ADJUSTABLE**

"Air-Out" valves, for eliminating air from steam radiators, operate by the expansion of a phosphor bronze diaphragm, soldered on the bottom of a brass float, containing a volatile liquid. This float is large in size, allowing ample capacity for the necessary pressure to operate the diaphragm.

The seating pin at the top of float is made from nickel silver, as called for in United States Government specifications for Air Valves. Verdigris will not form on nickel silver—corrosion at the vent port is therefore reduced to a minimum.



Inside Construction

One of the best valves money and experience can produce, and sold at a reasonable price.

Also made in straight pattern, threaded ⅛ in., ¼ in., and ¾ in. for quick venting of cellar mains and risers.

Guaranteed for One Year

Heavy Bronze Base
No Soldered Joints

We are making the "Air-Out" in a special straight Convector Type, for use on copper radiation. Specially adapted and guaranteed to work satisfactorily on one-pipe steam systems.

OTHER SPECIALTIES

- No. 10 Syphon Non-adjustable Air Valves.
- No. 476 Syphon Non-adjustable Air Valves.
- Carbon Post Automatic Air Valves.
- No. 10 Adjustable Floor and Ceiling Plates.
- Key and Wood Wheel Air Valves.
- Steam and Altitude Gauges.
- Water Gauges and Pop Safety Valves.

"Air-Out" Valves Guaranteed for One Year

Sarco Company, Inc.

MANUFACTURERS OF STEAM SPECIALTIES

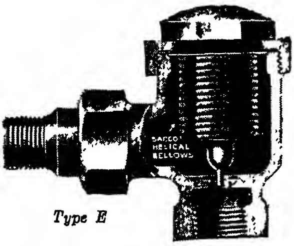
183 Madison Ave., **New York, N. Y.**

Branches in all Principal Cities

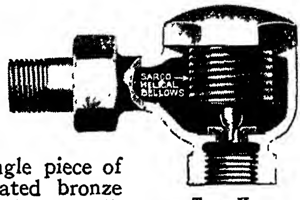
SARCO CANADA LIMITED, FEDERAL BLDG., TORONTO, ONT.

PRODUCTS—Sarco Radiator Traps, Steam Traps, Packless Inlet Valves, Air Eliminators, Alternating Receivers, Float Traps, Temperature Regulators, Self-Cleaning Strainers, Damper Regulators, Pipe Savers, Dial and Recording Thermometers.

SARCO RADIATOR TRAPS



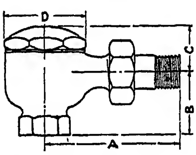
Type E



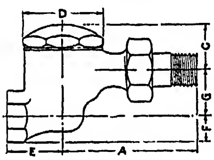
Type H

For vacuum, vapor and gravity heating systems, pressures up to 25 lbs. Available in angle, right and left offset and straightway patterns. Special features are the flexible bellows, made from a single piece of seamless, helically corrugated bronze tubing of large diameter and heavy wall. Also the self-aligning valve head which assures perfect seating. The valve has a full ¼″ lift, securing exceptionally large capacity. The body is of heavy brass, nickel plated with polished trimmings.

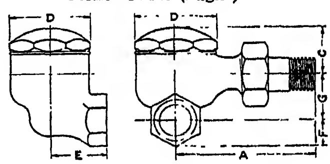
Angle



Straightway



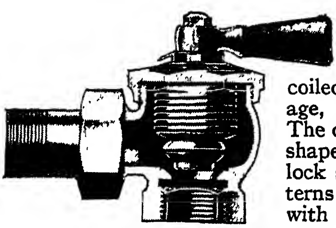
Corner-Offset (Right)



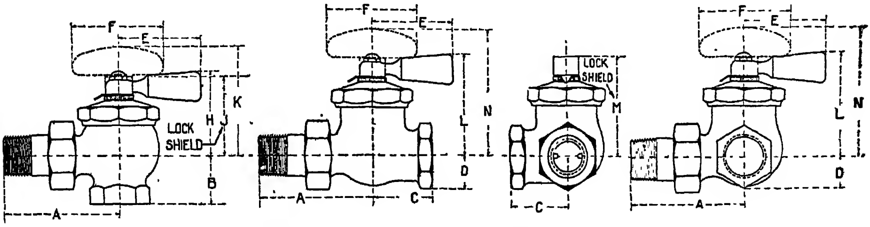
ROUGHING-IN DIMENSIONS

Type	Size	Dimensions							Capacity		List Price
		A	B	C	D	E	F	G	Vacuum	Vapor	
H	½″	3¼″	1½″	1 13/16″	2″	1⅜″	⅝″	1⅛″	250 sq. ft.	200 sq. ft.	$5.00
E	½″	3¼″	1½″	2⅛″	2¼″	1⅜″	¾″	1⅛″	250 sq. ft.	200 sq. ft.	6.00
E	¾″	3¼″	1½″	2⅛″	2¼″	1⅜″	¾″	1⅛″	800 sq. ft.	600 sq. ft.	8.00
E	1″	3½″	2″	2¾″	2½″	....	....	....	1800 sq. ft.	1500 sq. ft.	15.00

SARCO BELLOWS-PACKLESS VALVES



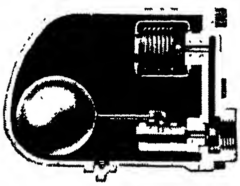
These valves are of the truly packless, quick-opening type. The valve stem is sealed to the cap by a flexible bellows made of the same seamless, helically coiled tubing, used in Sarco Traps. No possibility of leakage, binding or sticking—no attention or maintenance. The disc is of high temperature asbestos composition, cone shaped and fully protected. Lever or round handles, or lock shield furnished. Angle, straightway or corner patterns available. The body is of heavy brass, nickel plated with polished trimmings.



ROUGHING-IN DIMENSIONS

Size	Dimensions, Inches													List Prices	
	A	B	C	D	E	F	H	J	K	L	M	N	Capacities, Feet Direct Radiation	Angle	Offset or Straightway
1/2″	2 3/4	1 1/4	1 9/16	11/16	2 1/8	2 3/8	2 5/16	2	2 13/16	2 11/16	2 1/2	3 5/16	0–25	$5.50	$6.50
3/4″	3	1 1/4	1 9/16	13/16	2 1/8	2 3/8	2 5/16	2	2 13/16	2 11/16	2 1/2	3 5/16	26–75	6.00	7.00
1″	3 5/16	1 1/2	1 3/4	1	2 1/8	2 3/8	2 5/16	2 1/8	2 15/16	2 11/16	2 1/2	3 5/16	75–125	7.50	8.50
1 1/4″	3 1/2	1 5/8	...	...	2 1/8	2 3/8	2 5/8	2 7/16	3 1/16	...	....	....	126–200	9.50	10.50
1 1/2″	3 3/4	1 3/4	...	...	2 1/8	2 3/8	2 5/8	2 7/16	3 1/2	...	....	....	201–300	12.00	13.00

Capacities are for use with Vapor Systems. For Vacuum Systems, add 30 per cent.

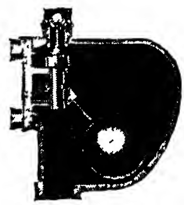


SARCO FT FLOAT AND THERMOSTATIC TRAPS

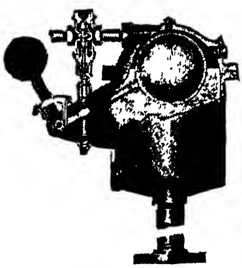
For dripping ends of mains, risers, stacks, unit heaters, hot water tanks, etc. Have internal thermostatic air by-pass. Made in three sizes, furnished with pipe connections 3/4″ to 2″ as shown below.

	FT-0, 3/4″	FT-0, 1″	FT-1, 1 1/4″	FT-1, 1 1/2″	FT-2, 2″
List Prices:	$16.00	$18.00	$20.00	$22.00	$60.00

SARCO AIR ELIMINATORS



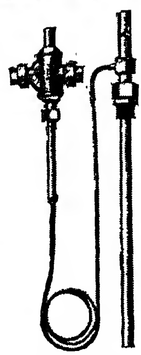
For venting air from Vapor Systems at one central point in basement. Has separate check valve to prevent air from entering system when operating below atmospheric pressure. Capacity sufficient to vent air from 15,000 sq. ft. of radiation in 20 minutes. 1″ I. P. S. List prices, No. 12 (illustrated) $25,00; No. 5A, 3/4″ for small systems, $9.00.



Alternating Receiver

SARCO ALTERNATING RECEIVER

For automatically returning condensate to boiler in Vapor Systems. Insures prompt return of water to boiler under all pressure conditions.



SARCO TEMPERATURE REGULATOR

A simple, self-contained automatic valve for regulating temperature of water in storage heaters, for room or kiln control. Made in sizes 1/2″ to 6″ for temperatures, 0–300°F.

SARCO STRAINERS

Strainers in pipe lines are effective insurance against injury to valves, traps, meters, etc. Suitable for steam, water and oil lines.

Write for Catalog HV-45.

The Trane Company

La Crosse, Wis.

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Also in 45 other cities—please consult your telephone directory

Trane Products

The many applications of Trane heating, cooling, ventilating, drying, processing and air-conditioning equipment make it impossible to discuss them all here. Because of this we are listing for the convenience of the engineer, the architect, and the contractor a list of available catalogs. Any or all of these catalogs will be sent on request to interested parties:

Heating Specialties

Complete catalog on steam and vacuum heating specialties which includes complete data on the Trane 14-corrugation bellows traps and other specialties.

Bulletin on orifice steam heating systems for residences and buildings requiring up to 800 square feet of radiation.

Temperature Control

Bulletin describing the Trane Temperature Control Valve with remote control and balanced pressure for controlling room temperatures.

Bulletin describing the Trane Temperature Control Valve for controlling vat, urn and liquid temperatures on special applications.

Convection Heaters

Color Bulletin containing complete story of Trane Convection Heaters.

Technical Data on Trane Convection Heaters.

Unit Heaters

Trane System of Unit Heating.

Unit Heater Data Bulletin.

Blast Heating and Special Drying and Processing Applications

Bulletin containing complete information on the applications of Trane extended heat transfer surface to fan systems of heating, drying, and process applications. Complete data.

Cooling

Trane Engineer's Data Book on Cooling.

How to Figure Trane Cooling Surface Applications.

Roughing-in dimensions and data on Trane Cooling.

Air-Conditioning

The Trane Climate Changer for residences—heating and cooling.

The oil and gas burning Climate Changer Data.

The Steam Climate Changer Data.

Also air-conditioning data for all types of buildings.

Ventilation

The Trane Air-o-lizer a complete heating, ventilating and temperature control unit for schoolroom ventilation.

Pumps

Trane Small Centrifugal Pumps.

WARREN WEBSTER & COMPANY

Pioneers of the Vacuum System of Steam Heating

Camden, N. J. **Branches in over 60 Cities**

—since 1888 *Webster* *Systems of Steam Heating*

Manufacturers of Improved Webster Systems of Steam Heating, Webster Central Controls, Webster System Radiation, Webster Series "78" Traps for "Process" Steam Pressures.

WEBSTER MODERATOR CONTROL "Control by the weather"

IMPROVED WEBSTER SYSTEMS—Low pressure, two-pipe systems of steam circulation employing Webster Supply Valves and Webster Return Traps at radiators, and Webster Drip and Heavy Duty Traps at drip points of mains and risers. Webster Dirt Strainers, Boiler Protectors, Lift Fittings, Expansion Joints, etc., are used where needed.

Return line operation may be either vacuum, gravity or "vapor." Webster Type "R" equipment consisting of Boiler Return Trap and Vent Trap is used with gravity or "vapor" systems.

Steam for an Improved Webster System may be supplied from a boiler, street mains, or other source. Either cast-iron or Webster System Radiation (concealed, non-ferrous type) may be employed.

Balanced Steam Distribution—Improved Webster Systems use *accurately-sized* metering orifices in radiator supply connections and, where required, at intermediate points in branch mains. The sizes of orifices are determined by the Company's representatives in accordance with the Company's exclusive methods. These orifices provide *balanced* or *equalized* distribution of steam to all radiators, a condition which is essential for satisfactory control of steam delivery from a central point.

Webster System Radiation—Concealed, non-ferrous type for use exclusively with Improved Webster Systems. Is unique in that it combines *in a single unit*, a light-weight heating element of high efficiency with an orificed radiator supply valve, a radiator trap, and supply and return piping connections. Metal enclosure is furnished for installation within the wall. Webster System Radiation and enclosures are so designed that the entire heating element can be quickly removed without damage to plaster or paint. Space requirement reduced to a minimum and installation greatly simplified.

Webster Electric Moderator Control—A single, centrally-located, gradual-

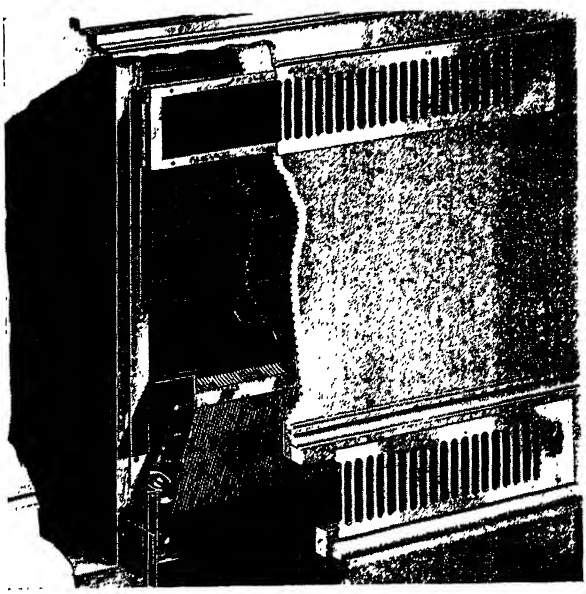


Fig. 1. Webster System Radiation

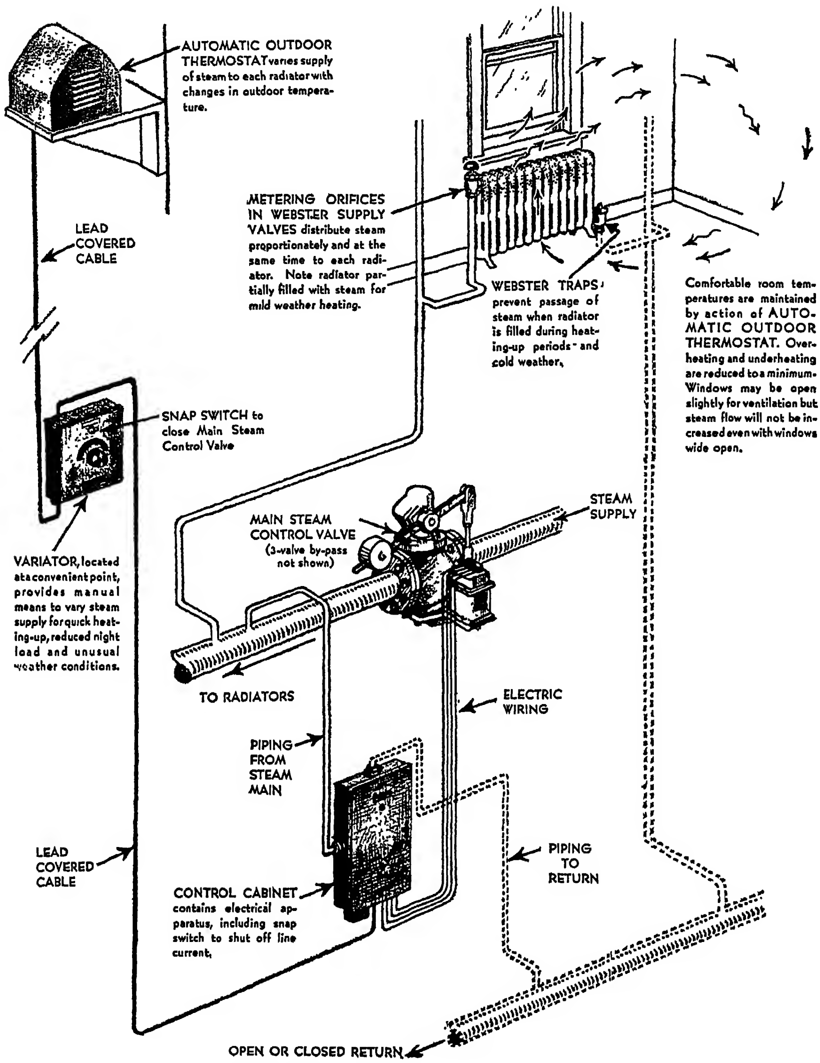


Fig. 2. Standard Arrangement of Improved Webster System with Webster Electric Moderator Control, Using a Single Main Steam Control Valve

acting control. Automatically varies amount of steam supplied to entire heating system in accordance with outside temperature. Heat delivery is continuous. Automatic Outdoor or "Roof" Thermostat operates Main Valve so that at minimum outdoor temperature (0° F. or −10° F., etc.) all radiators receive full heat. As outdoor temperature rises, steam delivery is proportionately reduced until at 70° F. outdoor temperature radiators are empty.

While largely automatic Moderator Control includes a manual Variator which permits operator to modify effect of Roof

Thermostat. Variator is used for heating-up, reduced night load, unusual weather conditions, etc.

Webster Moderator Control is provided in (1) a single Control Valve arrangement which adequately meets requirements of average building and (2) two or more Valve arrangement for zoning large installations or for groups of buildings. With the latter, each Valve is under the control of its own Outdoor Thermostat and each Valve is adjustable by means of an individual Variator.

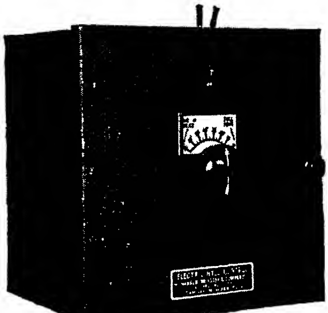


Fig. 3. Electric Hylo Controller

Application—For all types of Improved Webster Systems of 3,000 sq. ft. or more equivalent direct cast-iron radiation. *Particularly desirable for larger buildings.*

Webster Pneumatic-Type Moderator Control—Uses compressed air as medium of operation. Results in comfort and economy same as with the electric type. Applicable to large buildings, particularly zoned buildings or groups of buildings using two or more Main Control Valves operated by a single Roof Thermostat and Variator.

Webster Electric Hylo Control—A *manual* central control *especially applicable to two-pipe orifice systems using "street" steam.* Steam is supplied to each radiator in short pulsations of varying length. Length of pulsation is changed by moving knob on Hylo Controller according to changes in outdoor temperature. A rotating cam in cabinet opens and closes Control Valve in steam main. Provides continuous heating effect (no intermittent heat or "cold 70") with low cost central control for small and medium sized buildings.

Webster Hylo Steam Control—A *manual* central control of the graduated, continuous flow type to vary steam supply to entire system according to changes in outdoor temperature. Consists of Hylo Steam Variator (Fig. 4) and Type "E" Main Steam Control Valve as used with Moderator Control (see Fig. 1). Position of weights on beam scales causes motor to proportionately open or close Type "E" Valve in steam main. Ordinary variations in supply and return mains are compensated for automatically. Can be furnished with indoor thermostat for automatic overheat limit control and clock for automatic shut-off. Applicable to two-pipe orificed systems using low pressure steam from any source. Suitable in general for medium sized buildings.

Hylo Vacuum Variator Control—A *manual* central control of the graduated, continuous flow type *for vacuum systems only.* Operating principle similar to Hylo Steam Variator.

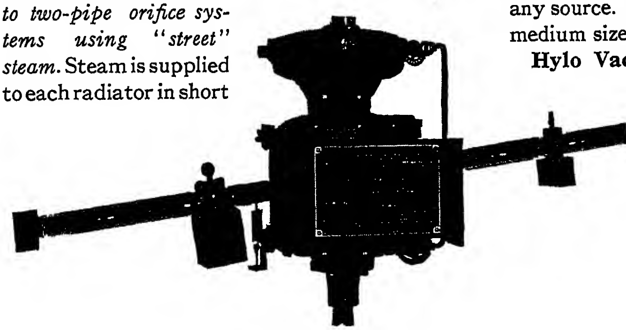


Fig. 4—The Webster Hylo Steam Variator

NOTE—Adjustable weights. Scale on beam is clearly marked reading from left to right "HEATING UP," "COLD WEATHER," "MODERATE WEATHER" and "MILD WEATHER," making adjustment simple and understandable by unskilled operators.

Wright-Austin Co.

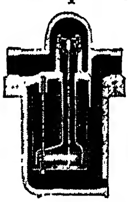
315 West Woodbridge Street
Detroit, Michigan

PRODUCTS—Steam Traps, Strainers, Air Traps, Steam and Oil Separators, Compressed Air Purifiers, Exhaust Heads, Pump Governors, Boiler Feed Water Regulators, Alarm Water Columns.

"Combination" Steam Trap

Made with internal thermostatic air by-pass and internal strainer. For vacuum service, also pressures up to 100 lbs. An excellent trap to take away condensation and air from heating apparatus.

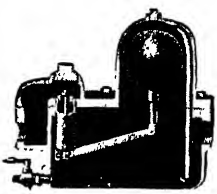
"Airxpel" Bucket Type Steam Trap



0 to 300 lbs.

This is a "double duty" trap expelling entrained air and condensate automatically. Unusually large capacity. Has exceptionally long life without repairs. All parts are accessible. Horizontal pipe connection; easy to install; hangs on the pipe line like a valve. Made in small sizes.

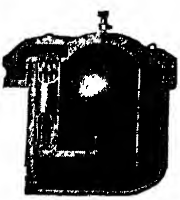
"Victor" Low Pressure Steam Trap



0 to 20 lbs.

For heavy volumes of condensation at low pressures. Furnished with thermostatic air by-pass for vacuum return lines. Makes a reliable, non-choking oil and grease trap, because of the large valve opening.

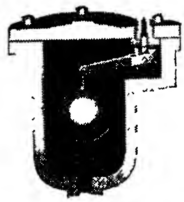
"Emergency" Float Type Steam Trap



0 to 250 lbs.

Three valve trap with large capacity at high loads and no wire drawing at low loads. No change of valves or adjustments from 0 to 200 lbs. Strong nickel float. An exceptionally reliable trap for use in inaccessible places.

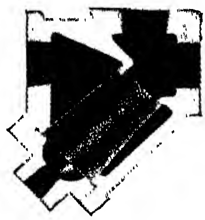
Air Trap for Relieving Air from Hot and Cold Water Systems



0 to 150 lbs.

Extreme simplicity and reliability characterizes this air trap. There is nothing to it but a float, a lever and a valve. No overflow needed, ample valve opening. Seven inches high by 6 inches diameter.

"Tuway" Angle or Straightway Strainer

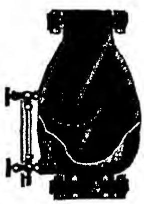


0 to 300 lbs.

An alternate inlet enables this strainer to be used either as an angle or straightway type in horizontal or vertical pipe line. The perforated brass screen has 400 holes per square inch but can be made to suit requirements.

Steam Separators

Type "A" Vertical Type "B" Horizontal



0 to 250 lbs.

Type "S" Horizontal, Self-Cleaning Oil Separator

0 to 60 lbs.

Will remove oil from exhaust steam down to a very small fraction of 1 per cent. Self-cleaning, cast in one piece requiring no maintenance. Thousands in use throughout the world.

See Also Page 776

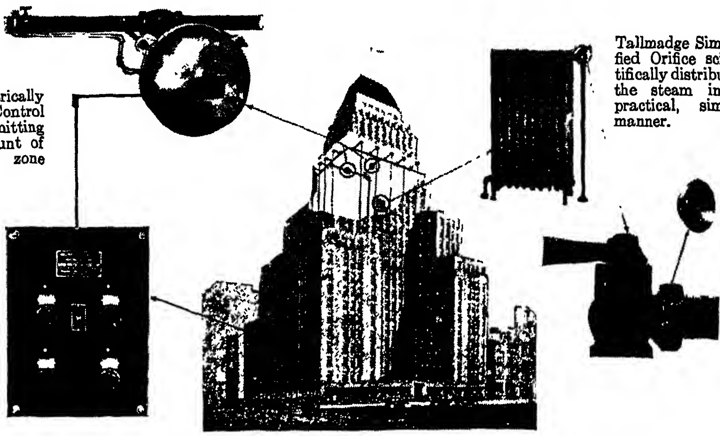
Webster Tallmadge & Co., Inc.

New York, N. Y.

Factory: East Orange, New Jersey

THE TALLMADGE ZONE HEATING SYSTEM

With Simplified Orifice Distribution and Compensated Control
Remote or Locally Controlled
Residence Heating Systems



Tallmadge electrically operated Zone Control Valve for admitting the proper amount of steam to each zone separately.

Tallmadge Simplified Orifice scientifically distributes the steam in a practical, simple manner.

Control Center of Tallmadge System. Can be placed anywhere.

Simplicity Predominates in the Tallmadge System of Zoned Heating
The devices illustrated above are the only ones used

The Tallmadge Zoned Heating System increases building comfort and saves the steam usually wasted in overheating in ordinary system operating without control.

In the application of the Tallmadge control the heating system is divided into separate systems or *zones,* each zone consisting of the radiators serving the same general building exposure or type of occupancy. A *control valve and mechanism* is placed on the steam supply to each zone and is electrically connected to a central control board. An *orifice* or restricted opening is placed in the entrance to each radiator. The *continuous* (*not intermittent*) *steam flow* to the radiators of a zone is then controlled (automatically or by hand from central control board) in accordance with the requirements of weather or occupancy affecting the steam demand of each zone which varies with relation to that of other zones and with every change in weather, thus saving additional steam on zones exposed to the sun or protected from the wind.

Radiator orifices are sized in proportion to the size of each radiator; fit in any make of radiator valve; *no orifices in mains or risers,* hence no unbalancing of steam distribution when radiators are changed or turned off.

The *valve control mechanism* automatically increases or decreases the continuous steam supply to the zone as radiators are turned on or off. It opens slowly to prevent noise usually resulting from too quick starting up and closes slowly to prevent blowing of the boiler safety valve; reduces peak demand on district steam supply; is motor-driven, taking current from the lighting circuit; operates on any initial pressure thus eliminating reducing valves and permitting smaller mains supplying steam to the control valve; prevents more than full steam flow entering any radiator hence *radiator traps* and *vacuum are unnecessary* regardless of the height or size of the building. Valve can be hand-operated. Operating mechanism can be removed without shutting off steam. No by-pass required.

No special pipe sizes required as those used in good standard practice for steam and return are satisfactory. Unnecessary to zone the returns as they may be common to all zones.

Low initial cost, low installation cost in new or old system due to simplicity of few devices used. Cost often offset by elimination of equipment otherwise necessary. Depreciation and maintenance practically nil due to few moving parts, hence efficiency does not decrease with age or use.

Julien P. Friez & Sons, Inc.,

(Subsidiary of Bendix Aviation Corporation)

Baltimore, Maryland, U. S. A.

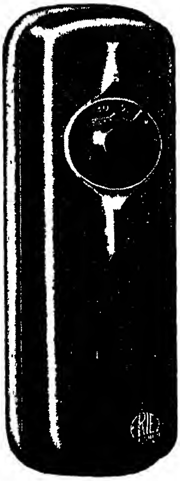
Established 1876

MANUFACTURERS OF METEOROLOGICAL, HYDROMETRIC AND AIR CONDITIONING INSTRUMENTS

PRODUCTS—Humidistats, Thermostats, Indicators, Thermometers, Hygro-thermographs, controlling and recording instruments for heating, ventilating and air conditioning—industrial, domestic and research fields.

Weather instruments, indicating and recording, all types.

Special remote indicating and recording equipment of various kinds.

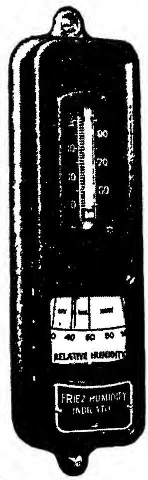


Humidistat—Extremely sensitive, accurate and reliable. Uses specially prepared, multiple human hair element. Setting graduated in percentage relative humidity over range from 10% to 100%. For use on 2 or 3 wire, high or low voltage electric circuits. Highest quality anodized finish in "bronze" or satin "silver" and suitable for all classes of comfort and industrial applications. The finest instrument of its type at attractive prices.—Apply for Bulletin "A."

Thermostat—First class, accurate and reliable instrument that matches exactly with Humidistat. 2 and 3 wire system types. Very finest finish and appearance.—Apply for Bulletin "E."

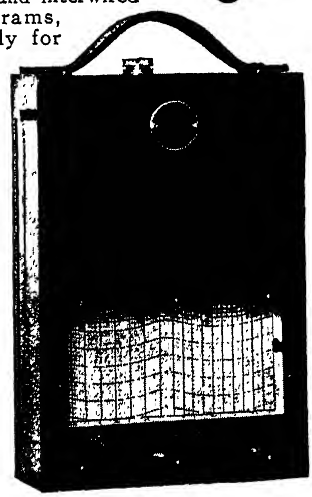
Effective Temperature Control—The new Friez "Comfortrol," in the one instrument, provides automatic control of—heating, cooling, humidifying and dehumidifying or any two or three of those four functions. Controls in terms of A.S.H.V.E. Comfort Chart Effective Temperatures and marks a great advance.—Apply for Bulletin "C."

New Friez Recorder—Specially designed for air conditioning and refrigeration work. Available for—Humidity, Temperature or Operation records or for any two or all three in the one instrument. Low in cost, accurate, reliable, compact, portable, rugged and at low prices. Of great value in all tests, surveys, research and as sales and service aid.—Apply for Bulletin "G."



Humidity and Temperature Indicator—Very accurate, sensitive, easy to read. Covers range 10% to 90% R.H. Human hair element. First class instrument for comfort, industrial and refrigeration work.—Apply for Bulletin "D."

Control Assemblies for Air Conditioning—We specialize in complete equipment, instruments, relays, transformers, switches, solenoid valves, dampers, etc., for automatic control of motors, fans, pumps, water or compressed air. Furnished assembled and interwired with diagrams, etc.—Apply for Bulletin "B."



Weather Instruments (all types—recording or indicating)—For 58 years we have been the leading builders of meteorological equipment for Government, Research and Industrial use. We have many applications of interest to engineers and architects for use in large buildings and homes, and for application to heating and ventilating control. See our equipment—Daily News or Empire State Building, New York.

We solicit your inquiries for all types of indicating, recording or controlling equipment.

Barber-Colman Company

Rockford, Illinois

BARBER-COLMAN ELECTRIC SYSTEM OF TEMPERATURE AND HUMIDITY CONTROL

Branches and Representatives

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The principal feature of the Barber-Colman *Electric* System of Temperature and Humidity Control is that it uses *electric*-motor-operated valves and damper controllers governed by *electrically* connected thermostats. Sufficient variety of equipment is available for the accurate and efficient control of any of the various types of heating, ventilating, or air conditioning installations. Special equipment and accessories are available to take care of special conditions. Competent engineering counsel is available without charge.

Thermostats and Hygrostat

Thermostats (and the Hygrostat) contain the sensitive elements of the control system—the elements that detect any change from the desired condition, and actuate the valves or damper controllers accordingly. The thermostats have bi-metal sensitive elements, the Hygrostat a belt of human hairs. Available types include the following:

Room Thermostat.
Two-Temperature Thermostat.
Duplex Thermostat.
Compound Thermostat.
Duct Thermostat.
Furnace Thermostat.
Air Stream Thermostat.
Immersion Thermostat.
Surface Thermostat.
Hygrostat.

Further information on request.

Motor-Operated Valves

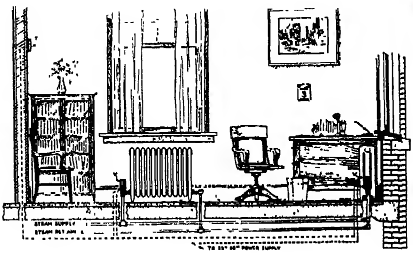
Motor-Operated Valves are used to control the flow of fluids, such as steam, water, gas, oil, etc., in pipes. Positive valves are for shut-off service where the control requirements call for either full flow or complete shut-off, whereas reversing valves provide regulating or throttling control. Both positive and reversing valves are available in standard body patterns, a wide range of sizes, and with operators to suit practically all requirements. Data sheets are available describing all types of valves and giving dimensions, electrical data, materials, lift of disc, etc. Principal types available are as follows:

Packless Valves ½ in. to 2 in.
Single Seat Packed Valves ¼ in. to 3 in.
Single Seat Packed Valves 2 in. to 6 in.
Pilot Piston Valves ½ in. to 8 in.
Semi-Balanced Vee-Port Valves ½ in. to 4 in.
Full-Balanced Vee-Port Valves 5 in. to 12 in.
Three-Way Valves ¼ in. to 4 in.
Butterfly Valves 2 in. to 16 in.

Damper Controllers

Motor-Operated Damper Controllers will open and close single or multi-louvred dampers in the ducts of unit ventilators, indirect warm air heating systems, blast systems, furnaces, ovens, etc. The controller is connected to the damper and positions it in accordance with indications from a thermostat or other switch. Most types of Damper Controllers are available with either positive (for open-and-shut operation) or reversing (for positioning operation) motors. Two standard sizes are made, as well as several specialized types. Complete information is provided in data sheet form.

EXAMPLES OF CONTROL SYSTEMS

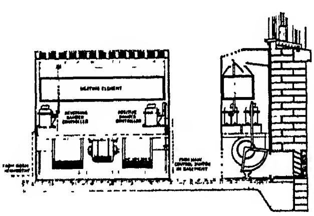


Direct Radiation — Motor - operated valves on the radiators are connected to and controlled by the thermostat on the wall. For a large room it is sometimes advisable to have several thermostats, each controlling a group of radiators.

With an automatically-controllable heat source—such as an oil or gas-fired boiler, a mechanical stoker, or a valve on a purchased-steam line—a "relay control" can be used. With an oil burner, for instance, the relay control will operate the burner and keep steam in the lines as long as one or more thermostats in any room or rooms is calling for heat. When all thermostats are satisfied, the burner is shut off. In this way, heat is always available whenever and wherever needed.

Any system for the control of direct radiation is easily installed on existing heating equipment.

Unit Ventilator—A conventionalized unit ventilator is illustrated. There are two damper controllers, one on the intake damper and one on the mixing damper. A room thermostat on the opposite wall actuates the mixing damper controller, positioning it to provide an output of warmer or cooler air, as required. The intake damper controller is governed by a manual switch in a central location (such as the janitor's office) and is positioned in accordance with weather conditions so that fresh outside or recirculated air may be taken in.

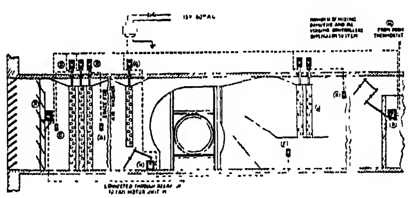


A motor-operated valve on the steam line may be used. Sometimes an airstream thermostat in the path of the output air is used as a pilot to prevent cold blasts.

Other types of controls—specialized valves, and the like—are made to suit the requirements of various designs of unit ventilators.

These controls may be purchased as original equipment on new unit ventilators or may be installed on existing heating systems using such units.

Blast System—A full blast heating system is diagrammed. Any part of it, if used separately, may be controlled approximately as shown.



An important element of this type of control system is the motor-operated valve (G) which is governed by the action of the damper controller (H) which, in turn, is controlled by the thermostat (F). The valve (G), which operates *slowly*, will remain stationary (it may be open, or closed, or in a throttling position) as long as the by-pass damper, which operates relatively fast, is "floated" by its controller (H) somewhere between the wide open and completely closed positions. In other words, the by-pass damper controller acts as a limit switch which, when the by-pass damper reaches its maximum cooling position, cuts off the steam supply to the reheating coil *slowly*, and, when the by-pass damper reaches the maximum heating position, opens the steam valve to the reheating coil *slowly*.

Variations of the controls shown can be adapted to any heating system of this general type.

Johnson Service Company

TEMPERATURE AND HUMIDITY CONTROL

General Offices and Factory

Milwaukee, Wis.

Branch Offices in all Large Cities

JOHNSON TEMPERATURE REGULATING CO. OF CANADA, LTD., 97 JARVIS STREET, TORONTO, ONT.
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Products and Services

Manufacturers, engineers, and contractors for Automatic Temperature and Humidity Control Systems applied to all types of *heating, ventilating, and air conditioning installations.*

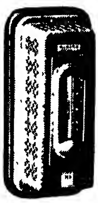
Temperature and Humidity Control for every range required in *manufacturing and industrial processes.*

Johnson "Duo-Stats" to maintain the proper relationship between outdoor and radiator temperatures.

Periodic Flush Systems for intermittent flushing in various sections of a building, reducing load on piping system and insuring economy in use of water.

Special bulletins and catalogues on request.

Johnson All-Metal Thermostats

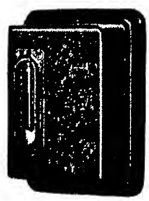


Room Thermostat

All Johnson thermostats, both room and insertion types, are *all-metal* throughout, having no soft or hard rubber parts to deteriorate and become inoperative. Every thermostat is precise in construction and is thoroughly tested for accuracy, efficiency, and durability.

The Johnson *intermediate action thermostat* gives a true graduated motion to mixing dampers and valves. It holds them in an intermediate position to maintain the temperature of the room accurately within one degree above or below the setting of the thermostat.

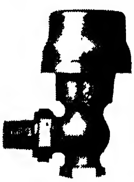
Johnson *diaphragm valves* are simple and rugged. Seamless metal bellows and heavy spring operate the valve stem. No complicated moving parts.



Dual Thermostat

Johnson Dual or Two-Temperature Thermostat

The *Dual, two-temperature, thermostat* is especially adapted for use where various rooms or groups of rooms are occupied when the remainder of the building is not in use. Separate steam mains are avoided. The shifting from "day" or occupancy temperature to an economy temperature for "non-occupancy" conditions, is accomplished by a switch or Johnson program clock at a central point. Push buttons on the thermostat are provided when "occupant control" is desired. Dual thermostats are all-metal and operate valves and dampers gradually to maintain temperature accurately within one degree.



Sylphon Valve

Humidostats and Humidifiers

The *Johnson humidostat* automatically controls the supply of moisture delivered to the air by a humidifier or air washer and maintains a constant percentage of relative humidity. Available in both room and insertion patterns and with elements as determined by requirements, the most sensitive controlling within 1 per cent at relative humidity of 95 per cent for 100 degrees F.

Johnson humidifiers are furnished in steam "grid" type or pan type with copper evaporating pan, brass heating coil, and float control.

Room Humidostat (Cover removed)

Ferro-Nil Corporation

500 Fifth Avenue **New York City**

WATER CONDITIONING FOR AIR CONDITIONING

Photograph reprinted with permission of National Broadcasting Company

All water used for washing air picks up corrosive matter from the air. The corrosive matter may be oxygen, carbon dioxide, sulphur tri-oxide or sulphur di-oxide. In air conditioning systems where water is recirculated, the amount of acidic material contained in the water may reach very high levels. This acid water will corrode metallic surfaces with which it comes in contact. In severe cases equipment has been known to fail after three months operation due to corrosive action of water used.

Ferro-Nil Service by maintaining water in a non-corrosive condition, prevents corrosion. Corrosion[1] causes high rates of depreciation. Ferro-Nil Service by maintaining metallic surfaces free from rust and scale affords optimum heat transfer efficiencies at all times.

We number among our clients, nationally known corporations such as:—

Rockefeller Center
National Broadcasting Company
Columbia Broadcasting Company
Runkel Chocolate Company
Metropolitan Life Insurance Company
U. S. Government

SUBSTANTIAL DOLLAR SAVINGS GUARANTEED

Let us submit estimate of cost of Ferro-Nil Service for your air conditioning equipment.

The illustration depicts a typical installation used on dehumidifier by the National Broadcasting Company. Note simplicity of design with consequent simplicity of installation and operation.

[1]See discussion of Corrosion Chapter 34.

Index to Modern Equipment

Numerals following Manufacturers' Names refer to pages in the Catalog Data Section

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AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS GUIDE, 1934

Roll of Membership

AMERICAN SOCIETY *of*
HEATING *and* VENTILATING ENGINEERS

1934

Contains Lists of Members Arranged Alphabetically and Geographically, also Lists of Officers and Committees, Past Officers and Local Chapter Officers

Corrected to January 1, 1934
Published at the Headquarters of the Society
51 Madison Avenue, New York, N. Y.

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American Society *of* Heating *and* Ventilating Engineers

51 Madison Ave., New York, N. Y.

1933-34

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1933-34

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A

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ABEL, D. Morgan (*J* 1928), Box 21, Rochelle, La.

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ADAMS, Neil D. (*M* 1929; *A* 1925; *J* 1922), Supt. (for mail), Franklin Heating Sta., 220 Second Ave. S.W., and 836 Eighth Ave. S.W., Rochester, Minn.

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AKERS, George W. (*M* 1929), Secy-Treas. (for mail), George W. Akers Co., 2847 Grand River Ave., Detroit, and 424 Willitts, Birmingham, Mich.

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ALVORD, Arthur M. (*M* 1926), Pres. (for mail), Alvord & Swift, Grand Central Terminal, New York, and 240 Hamilton Ave., New Rochelle, N. Y.

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BAKER, Howard C. (*M* 1921), The H. C. Baker Co., 128 S. St. Clair St., Toledo, Ohio.
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BAKER, Roland H. (*M* 1928; *A* 1924), Pres. (for mail), Baker Engrg. Corp., 145 Broadway, and 244 Brattle St., Cambridge, Mass.
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BALSAM, Charles P. (*M* 1932), Sales Director, House Htg. Div. (for mail), Peoples Gas Light Co., and Illinois Athletic Club, Chicago, Ill.
BAMPTON, C. Morton (*M* 1919), Gates Htg. Co., Inc., 915 Gates Ave., Brooklyn, N. Y.
BARBERA, Henry A. (*S* 1932), Student, New York University, College of Engrg., New York, and (for mail), 3146–102nd St., Corona, L. I., N. Y.
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BARRY, James G., Jr. (*M* 1933), Vice-Pres. (for mail), Elliott & Barry Engrg. Co., 4060 W. Pine Blvd., and 5051 Queens Ave., St. Louis, Mo.
BARRY, Patrick I. (*M* 1920), M. Barry, Ltd., 4 Marlboro St., Cork, Ireland.
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BARTLETT, C. Edwin (*M* 1922), Pres. (for mail), Bartlett & Co., Inc., 1938 Market St., and 3111 W. Coulter St., Philadelphia, Pa.
BASTEDO, Albert E. (*M* 1919), Vice-Pres-Treas-Mgr. (for mail), Burnham Boiler Corp., Irvington-on-Hudson, and Burnside Dr., Hastings-on-Hudson, N. Y.
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BERNHARD, George (*A* 1929), Pres., Bernhard Engrg. Corp., 101 Park Ave., New York, and (for mail), 18 Lismore Rd., Lawrence, L. I., N. Y.
BERNSTROM, Bert (*M* 1930), Engr., Lakeside Co., Hermansville, Mich.
BEST, Millard W. (*A* 1933), Pres. (for mail), Kolelectric Underfeed Stoker Co., Ltd., 245 Kenilworth Ave. S., and 1750 King St. E., Hamilton, Ont., Canada.
BETTS, Howard M. (*M* 1927), Senior Mech. Engr., Htg. & Vtg. (for mail), Dept. of Bldgs., City of Minneapolis, 213 City Hall, and 4923 Russell Ave., S., Minneapolis, Minn.
BETZ, Harry D. (*M* 1928), Pres. (for mail), Betz Unit Air Cooler Co., 6 W. Ninth St., and 4210 Mercer, Kansas City, Mo.
BINDER, Charles G. (*M* 1920), Mgr. Htg. Dept., Warren Webster & Co., 17th and Federal Sts., Camden, and (for mail), 115 Oak Ter., Merchantville, N. J.
BINFORD, Wilmer M. (*J* 1930), Mgr. Contract Dept., So. Div. (for mail), 2120 East 25th St., and 6215 San Vicente Blvd., Los Angeles, Calif.
BIRCH, Herbert R. (*M* 1922), U. S. Radiator Corp., 370 Lexington Ave., New York, N. Y.
BIRKHOLZ, H. E. (*A* 1925), Novelaire Corp., 1114 Bardstown Rd., Louisville, Ky.
BIRRELL, Allan L. (*A* 1925), Consulting Engr. (for mail), 372 Bay St., Toronto 2, and 93 Kingsway, Old Mill P. O., Ont., Canada.
BIRUKOFF, Roman R. (*S* 1933), 2034 Grand Concourse Apt. 4c, New York, N. Y.
BISCH, Bernard J. (*M* 1931), Engr., St. Mary of The Woods College, St. Mary of the Woods, Ind.
BISHOP, Charles R. (*Life Member; M* 1901), 413 Locust St., Lockport, N. Y.
BISHOP, Frederick R. (*M* 1921), Mfrs. Agt., 8836 Quincy Ave., Detroit, Mich.
BJERKEN, Maurice H. (*A* 1927), Dist. Repr. (for mail), Hoffman Specialty Co., 533 S. Seventh St., and 4952–17th Ave. S., Minneapolis, Minn.
BLACK, Edgar N., 3rd (*M* 1922), Philadelphia Mgr., Fitzgibbons Boiler Co., Inc., 814 Land Title Bldg., Philadelphia, and (for mail), 111 Woodside Rd., Haverford, Montgomery Co., Pa.
BLACK, F. C. (*M* 1919), Pres. (for mail), F. C. Black Co., 622 W. Randolph St., and 4535 N. Ashland Ave., Chicago, Ill.
BLACK, George E. (*M* 1915), 709 Broad St., Sewickley, Pa.
BLACK, Harry G. (*M* 1917), Prop. (for mail), P. Gormly Co., 155 N. Tenth St., and 927 North 65th St., Philadelphia, Pa.
BLACK, William B. (*J* 1932), Salesman, Bryant Heater Co., 2341 Carnegie Blvd., Cleveland, Ohio.
BLACKBURN, Edwin C., Jr. (*M* 1929), Consulting Engr., 12 Clermont Ave., Hempstead, L. I., N. Y.
BLACKHALL, Wilmot R. (*M* 1922), 332 Waverley Rd., Toronto, Ont., Canada.
BLACKMAN, Alfred O. (*M* 1911), Consulting Engr. (for mail), 145 West 45th St., and 149 West 12th St., New York, N. Y.
BLACKMORE, F. H. (*M* 1923), Mgr. Operating Dept. (for mail), U. S. Radiator Corp., Box 686, Detroit, and 515 Tooting Lane, Birmingham, Mich.
BLACKMORE, George C. (*Charter Member; Life Member*), Edgewood, Pittsburgh, Pa.

BLACKMORE, J. J.* (*Charter Member; Life Member*), 32 West 40th St., New York, N. Y.
BLACKMORE, James S. (*J*1931), 301 Brushton Ave., Pittsburgh, Pa.
BLACKSHAW, J. L.* (*J* 1929), Research Engr., A.S.H.V.E. Research Lab., 4800 Forbes St., and (for mail), 3728 Dawson St., Pittsburgh, Pa.
BLAKE, Albert Henry (*M* 1926), 4 Main St. S., Weston, Ont., Canada.
BLANDING, George H. (*M* 1919), 800 N. Lombard Ave., Oak Park, Ill.
BLANKIN, Merrill F. (*M* 1927; *A* 1926; *J* 1919), Pres. (for mail), Haynes Selling Co., Inc., 1518 Fairmount Ave., and 3328 W. Penn St., Philadelphia, Pa.
BLISS, George L. (*A* 1933), Engr. and Sales. (for mail), Allis-Chalmers Mfg. Co., 1410 Waldheim Bldg., 11th and Main, and 7641 Brooklyn Ave., Kansas City, Mo.
BLITZ, Emmanuel (*M* 1931), 2158 Crotona Ave., New York, N. Y.
BOALES, William G. (*A* 1923), (for mail), 6537 Hamilton Ave., Detroit, and 195 McMillan Rd., Grosse Pointe Farms, Mich.
BOCK, Bernard (*A* 1929; *J* 1927), Engrg. Draftsman, 425 Beech St., Arlington, N. J.
BODDINGTON, William P. (*M* 1927), Mgr. (for mail), The Canadian Powers Regulator Co., Ltd., 106 Lombard St., and 280 Clendenan Ave., Toronto, Ont., Canada.
BODINGER, Jacob H. (*M* 1931), 439 West 38th St., New York, N. Y.
BOGATY, Hermann S. (*M* 1921), 5230 North 15th St., Philadelphia, Pa.
BOLSINGER, Raymon C. (*M* 1916), Pres. (for mail), Automatic Florzone Htg. Co., Conshohocken, Pa., and 238 E. Madison Ave., Collingswood, N. J.
BOLTE, E. Endicott (*A* 1929), Salesman, Natl. Radiator Corp., 601, No. 1 N. LaSalle St., and (for mail), 6516 Kenwood Ave., Chicago, Ill.
BOLTON, Reginald Pelham* (*Life Member; M* 1897), (*Presidential Member*), (Pres., 1911; 1st Vice-Pres., 1905-1910; 2nd Vice-Pres., 1903; Bd. of Governors, 1901, 1905, 1910, 1911, 1912, 1913), The R. P. Bolton Co., 116 East 19th St., New York, N. Y.
BOND, Horace A. (*M* 1930), 12 Ramsey Pl., Albany, N. Y.
BOON, George (*M* 1915), Boon & Sample, Inc., 3008 Ludlow St., Philadelphia, Pa.
BOOTH, C. A. (*M* 1917), Vice-Pres. (for mail), Buffalo Forge Co., 490 Broadway, and 142 Summit Ave., Buffalo, N. Y.
BOOTH, Harry N. (*M* 1924; *A* 1917), Vice-Pres., Sales Dept. (for mail), U. S. Radiator Corp., Room 1056, 1st Natl. Bk. Bldg., and 688 Taylor Ave., Detroit, Mich.
BORNEMANN, Walter A. (*M* 1924; *J* 1923), Carrier-York Corp., 1541 Sansom St., Philadelphia, Pa.
BOSTAIN, James C. (*M* 1923), 3469 Evanston Ave., Cincinnati, Ohio.
BOUEY, Angus J. (*J* 1930), 706 Hollingsworth Bldg., Los Angeles, Calif.
BOUILLON, Lincoln (*M* 1933), Consulting Engr., 1411 Fourth Ave., Bldg., and (for mail), 4186-42nd Ave. N.E., Seattle, Wash.
BOWERS, Arthur F. (*A* 1919), Pres., Industrial Htg. & Engrg. Co., 828 N. Broadway, Milwaukee, Wis.
BOWERS, J. S. (*M* 1921), J. Sylvan Bowers Sales Co., 3805 Page Blvd., St. Louis, Mo.
BOWERS, Ross C. (*A* 1932), Br. Mgr. (for mail), Minneapolis-Honeywell Regulator Co., 335 W. North Ave., and 3773 North 52nd St., Pittsburgh, Pa.
BOWLES, Potter (*A* 1928), Pres. (for mail), Hoffman Specialty Co., 3707 Chrysler Bldg., New York, and 678 Ely Ave., Pelham Manor, N. Y.
BOYDEN, Davis S.* (*M* 1909), (Council, 1917-1930-1933-; Treas., 1933), Supt. Steam Htg. Service Dept. (for mail), Edison Electric Illuminating Co. of Boston, 39 Boylston St., Boston, and 1496 Commonwealth Ave., Brighton, Mass.
BOYNTON, Daniel W. (*A* 1927), International Heater Co., 77 Franklin St., Boston, Mass.
BRAATZ, Chester Johnson* (*M* 1930), Engr. Temp. Control, Barber-Colman Co., and (for mail), 1931 Douglas St., Rockford, Ill.
BRABBÉE, Charles W.* (*M* 1925), American Radiator Co., 675 Bronx River Rd., Yonkers, N. Y.
BRACKEN, John Henry (*M* 1927), Mgr., Industrial Uses Dept. (for mail), The Celotex Co., 919 N. Michigan Ave., Chicago, Ill.
BRADFIELD, William W. (*M* 1926), Consulting Engr. (for mail), 901 Michigan Trust Bldg., and 1352 Franklin St. S.E., Grand Rapids, Mich.
BRADFORD, H. H. (*A* 1927), Asst. Br. Mgr., Minneapolis-Honeywell Co., 2747 Fourth Ave. S., and (for mail), 4701 Blaisdell Ave., Minneapolis, Minn.
BRADLEY, Eugene P. (*M* 1906), Pres. (for mail), Hester-Bradley Co., 2835 Washington Ave., and 6935 Pershing Ave., St. Louis, Mo.
BRAEMER, William G. R. (*M* 1915), (for mail), Wm. G. R. Braemer and Josiah H. Smith Engrs., Room 1265 Commercial Trust Bldg., Philadelphia, Pa., and Haddonfield, N. J.
BRAKENRIDGE, Charles E. (*A* 1930), 3817 Johnson Ave., Western Springs, Ill.
BRANDI, O. H. (*M* 1930), Office Mgr., Carrier Lufttechnische Gesellschaft, Berlin-Charlottenburg II, Savignyplatz 31 and (for mail), Berlin-Wilmersdorf Hohenzollerndamm 35, Germany.
BRANDT, Ernest Hamilton, Jr. (*M* 1928), Pres. (for mail), Reliance Engrg. Co., Inc., 1219 Eye St. N.W., Washington, D. C., and Frederick Apts., Charlotte, N. C.
BRASSINGTON, Arthur F. (*A* 1918), (for mail), 505-7 West 45th St., New York City, and 337 Richmond Ave., Port Richmond, N. Y.
BRAUER, Roy (*M* 1926), Pres. (for mail), Ventilating Equipment Corp., 1101 Bessemer Bldg., Pittsburgh, and R. F. D. No. 1, Hillcrest Library, Pa.
BRAUN, John J. (*M* 1932), Factory Mgr., The U. S. Playing Card Co., Norwood Sta., Cincinnati, and (for mail), 4305 Floral Ave., Norwood, Ohio.
BRAUN, Louis T. (*M* 1921), Executive Secy. (for mail), Chicago Master Steamfitters Assn., 228 N. LaSalle St., and 1548 Pratt Blvd., Chicago, Ill.
BRAYTON, William M. (*M* 1926), Sales Engr. (for mail), Robert Gordon, Inc., 22 W. Austin Ave., Chicago, and Cary, Ill.
BRECKENRIDGE, L. P.* (*Life Member; M* 1920), The Brackens, N. Ferrisburg, Vt.
BREDESEN, Bernhard P. (*A* 1931), 3119 Knox Ave. N., Minneapolis, Minn.
BREITENBACH, George C. (*M* 1933; *A* 1933; *J* 1928), Sales Engr. (for mail), The Trane Co., 2006 Chestnut St., Philadelphia, and 300 Essex Ave., Apt. 203A, Narberth, Pa.
BREMSER, Harry A. (*A* 1930), 230 W. Los Flores Dr., Altadena, Calif.
BRENEMAN, Robert B. (*A* 1931; *J* 1927), Sales Engr. (for mail), Armstrong Cork & Insulation Co., 232 W. Seventh St., and 1557 Addingham Pl., Cincinnati, Ohio.
BREWER, John G. (*S* 1933), 5549 Bryant St., Pittsburgh, Pa.
BRIDE, William T. (*M* 1928; *A* 1928; *J* 1925), Supt. Engrg. (for mail), P. O. Box 777, Lawrence, and 50 High St., Methuen, Mass.
BRIDGES, Frank G. (*M* 1919), 13602 McElhattan Ave., College Sta., Cleveland, Ohio.
BRIGHAM, Frederick H. (*M* 1930), Sales Engr., G. H. Gleason & Co., 25 Huntington Ave., Boston, and (for mail), 80 Bedford St., Lexington, Mass.
BRILL, Joseph W. (*M* 1932; *A* 1932; *J* 1931), 105 Canisteo St., Hornell, N. Y.
BRINTON, Joseph Ward (*M* 1920), Dist. Mgr. (for mail), American Blower Corp., 1003 Statler Bldg., Boston, and 51 Gleason St., West Medford, Mass.
BRISSETTE, Leo A. (*M* 1930), Treas. (for mail), Trask Htg. Co., 4 Merrimac St., Boston, and 168 Florence St., Melrose, Mass.

BRODERICK, Edwin L. (*M* 1933), Research Asst. in M. E. (for mail), University of Illinois, 105 M. E. Laboratory, and 1108 W. Stoughton St., Urbana, Ill.
BRONSON, Carlos E.* (*M* 1919), Mech. Engr. (for mail), Kewanee Boiler Corp., and 311 McKinley Ave., Kewanee, Ill.
BROOM, Benjamin A. (*M* 1914), Sales Promotion Engr., Weil-McLain Co., 641 W. Lake St., and (for mail), 1311 Farwell Ave., Chicago, Ill.
BROWN, Alfred P. (*M* 1927), B. F. Reynolds & Co., 609 N. LaSalle St., Chicago, Ill.
BROWN, Aubrey I.* (*M* 1923), Associate Prof. of Htg. and Vtg. (for mail), Ohio State University, and 169 Richards Rd., Columbus, Ohio.
BROWN, Foskett* (*M* 1926), Vice-Pres. (for mail), Gray & Dudley Co., 222 Third Ave. N., P. O. Box 722, and 2314 West End Ave., Nashville, Tenn.
BROWN, Morris (*J* 1928), 609 W. Park St., Dorchester, Mass.
BROWN, Ronald F. (*S* 1933), 66 Mitchell Ave., Binghamton, N. Y.
BROWN, Tom (*M* 1930), Owner Air Cond., Mfgrs. Repr. (for mail), 487 W. Alexandrine Ave., Detroit, and 40 Connecticut Ave., Highland Park, Mich.
BROWN, Thomas (*A* 1931), 41 Glenholme Ave., Toronto, Ont., Canada.
BROWN, William A. (*M* 1930), Consulting Engr. (for mail), Cucci & Brown, 347 Madison Ave., New York, and 929 East 28th St., Brooklyn, N. Y.
BROWN, William H. (*A* 1923), 3015 North 22nd St., Milwaukee, Wis.
BROWN, W. Maynard (*A* 1930), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.
BROWN, W. Murray (*S* 1930), Draftsman and Estimator (for mail), William P. Brown, 31 Sanford St., and 78 Randolph St., Springfield, Mass.
BROWNE, Alfred L. (*M* 1923), Illinois Engrg. Co., 3514 Grand Central Terminal, New York, N. Y.
BRUCKMANN, John C. (*S* 1932), 2290 Sedgwick Ave., New York, N. Y.
BRUEGGEMAN, Arthur R. (*M* 1920), A. R. Brueggeman Co., 1740 East 12th St., Cleveland, Ohio.
BRUNETT, Adrian L. (*M* 1923), Assoc. Mech. Engr., U. S. Supervising Architects Office, Treasury Dept., Washington, D. C., and (for mail), P. O. Box 236, Rockville, Md.
BRUNT, T. Bayard (*M* 1917), 405 Eighth St., Riverton, N. J.
BRUST, Otto (*M* 1930), Prague II, Revolucni 13, Czechoslovakia.
BRYANT, Dr. Alice G. (*M* 1921), 502 Beacon St., Boston, Mass.
BRYANT, Percy J. (*M* 1915), Chief Engr. (for mail), Prudential Insurance Co., 783 Broad St., Newark, and 754 Belvidere Ave., Westfield, N. J.
BUCHER, Harry G. (*J* 1930), Junior Htg. Engr., C. A. Dunham Co., 3002 Grant Bldg., Pittsburgh, and (for mail), 210 Copeland St., McKees Rocks, Pa.
BUCK, Lucien (*M* 1928), Pres. (for mail), Buck Dryer Corp., P. O. Box 308, Manchester, Conn.
BUCKLEY, Martin B. (*A* 1930), 910 Grand Ave., Kansas City, Mo.
BUENGER, Albert* (*M* 1920; *A* 1917), Mech. Engr. (for mail), C. H. Johnston Archt., 715 Empire Bk. Bldg., and 1606 Stanford Ave., St. Paul, Minn.
BUENSOD, Alfred Charles (*M* 1918), Sales Engr., Carrier Engrg. Corp., Chrysler Bldg., and (for mail), 1 Fifth Ave., New York, N. Y.
BUFORD, Jack W. (*S* 1933), 3 Hammond St., Cambridge, Mass.
BULKELEY, Claude A.* (*M* 1923), Chief Engr. (for mail), Niagara Blower Co., 6 East 45th St., and 25 Prospect Pl., New York, N. Y.
BULL, Alvah Stanley (*S* 1933), 304 West 35th St., Minneapolis, Minn.
BULLEIT, Charles R. (*M* 1932; *A* 1932; *J* 1930), 1630 S. Grand Ave., Evansville, Ind.
BULLOCK, Howard H. (*A* 1933), General Electric Co., 5201 Santa Fe Ave., Los Angeles, Calif.
BULLOCK, Thomas A. (*M* 1930), Engr. (for mail), Densmore, LeClear & Robbins, 31 St. James Ave., Boston, and 39 Fairmont St., Arlington, Mass.
BUNKER, Kenneth S. (*A* 1930), Sales Engr. (for mail), Union Tank & Pipe Co., 2801 Santa Fe Ave., and 352 N. Stanley Ave., Los Angeles, Calif.
BUR, Julien R. C. (*J* 1931), Chief Engr. (for mail), Bur & Co., 10 Rue du Chapeau Rouge, and 1 Place Francois Rude, Dijon, France.
BURBAUM, W. Allen (*J* 1933), Asst. Br. Mgr., Rex Cole, Inc., 2392 Grand Concourse, New York, and (for mail), 180 Clinton Ave., Brooklyn, N. Y.
BURDOIN, Allen J. (*S* 1933), 2 Sherman St., Lexington, Mass.
BURKE, Fletcher H. (*M* 1925), 677 Ellicott Sq., Buffalo, N. Y.
BURKE, James J. (*J* 1930), Engr., Carrier Engrg. Corp., 12 South 12th St., Philadelphia, and (for mail), 720 N. Broad St., Elizabeth, N. J.
BURNAP, Charles W. (*M* 1922), Herman Nelson Corp., 724 Commercial St., Emporia, Kan.
BURNETT, Earle S. (*M* 1920), Mech. Engr., U. S. Bureau of Mines, Amarillo Helium Plant, and (for mail), 4223 West 11th Ave., Amarillo, Texas.
BURNS, Edward J. (*M* 1923), Supt., H. Kelly & Co., 948 N. W. Bank Bldg., and (for mail), 4716 Aldrich Ave. S., Minneapolis, Minn.
BURNS, John R. (*S* 1933), 5035 Forbes St., Pittsburgh, Pa.
BURRITT, Charles G. (*A* 1916), Mgr. Minneapolis Office (for mail), Johnson Service Co., 922 Second Ave. S., and Buckingham Hotel, Minneapolis, Minn.
BUSHNELL, Carl D. (*A* 1921), Pres. (for mail), The Bushnell Machinery Co., 1501 Grant Bldg., Pittsburgh, and 94 Pilgrim Rd., Rosslyn Farms, Carnegie, Pa.
BUTLER, Peter D. (*M* 1922), Salesman, U. S. Radiator Corp., Davis and Central Ave., Harrison, and (for mail), 127 Edgewater Rd., Grantwood, N. J.
BUTT, Roderick E. W. (*J* 1930), 3 Orme Court, London W2, England.
BYNUM, Otis W. (*A* 1930), Carrier Engrg. Corp., 850 Frelinghuysen Ave., Newark, N. J.

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CALDWELL, Arthur C. (*M* 1930), 550 South 48th St., Philadelphia, Pa.
CALEB, David T. (*M* 1923), Engr. (for mail), Kansas City Power & Light Co., 1330 Baltimore Ave., and 141 Spruce St., Kansas City, Mo.
CALLAGHAN, Philip F., Jr. (*J* 1929), Sales Mgr., D. G. C. Trap & Valve Co., 9 East 46th St., New York, and (for mail), 3003 Ave. I., Brooklyn, N. Y.
CALVERT, Norman W.* (*M* 1921), 20 Lime St., Boston, Mass.
CAMPBELL, Alfred Q., Jr. (*J* 1933), Salesman, E. K. Campbell Htg. & Vtg. Co., and (for mail), 1083 Meriwether Ave., Memphis, Tenn.
CAMPBELL, Everett K.* (*M* 1920), (Council, 1931-1933), Pres. and Treas. (for mail), E. K. Campbell Htg. Co., 2445 Charlotte St., and 3717 Harrison Blvd., Kansas City, Mo.
CAMPBELL, E. K., Jr. (*J* 1930), Thermidaire Corp., 2445 Charlotte St., Kansas City, Mo.
CAMPBELL, F. B. (*A* 1927), (for mail), American Radiator Co., 40 West 40th St., New York, and 245 Macon St., Brooklyn, N. Y.
CAMPBELL, Thomas F. (*M* 1928), Minneapolis-Honeywell Regulator Co., 1013 Penn Ave., Wilkinsburg, Pa.
CAMPBELL, Walter E. (*M* 1931), 619 Milan Ave., South Pasadena, Calif.
CANDEE, Bertram C. (*M* 1933), Partner, Beman & Candee, 374 Delaware Ave., Buffalo, and (for mail), 19 Tremont Ave., Kenmore, N. Y.

CANNON, C. Newton (*S* 1933), General Electric Co., and (for mail), 13 State St., Schenectady, N. Y.
CAREY, James A. (*M* 1928), Carrier Engrg. Corp., Newark, N. J., and (for mail), Villanova, Pa.
CAREY, Paul C. (*M* 1930), (for mail), Runyon & Carey, 33 Fulton St., Newark, and 31 Claremont Dr., Maplewood, N. J.
CAREY, Thomas M. (*M* 1931), 622 Jefferson Ave., Defiance, Ohio.
CARLE, William E. (*M* 1926), Pres. (for mail), Carle-Boehling, Co., Inc., 1641 W. Broad St., and 2220 Floyd Ave., Richmond, Va.
CARLSON, Everett E. (*M* 1932, *A* 1929), Br. Mgr. (for mail), The Powers Regulator Co., 1010 Louderman Bldg., and 6652 Washington Ave., St. Louis, Mo.
CARMAN, George G. (*A* 1931; *J* 1928), Lewis Institute, Chicago, Ill.
CARPENTER, R. H. (*M* 1921), (Council, 1930-1933), Mgr., New York Office (for mail), Nash Engrg. Co., Graybar Bldg., 420 Lexington Ave., New York, and 10 Jefferson Ave., White Plains, N. Y.
CARR, Maurice L. (*M* 1931), Director (for mail), Pittsburgh Testing Lab., P. O. Box 1115, and Webster Hall, Pittsburgh, Pa.
CARRASCO, Saturnino (*A* 1932; *J* 1928), Monedo 944, Casilla 2405, Santiago, Chile, S. A.
CARRIER, Earl G. (*J* 1929), Estimating Engr., Carrier Engrg. Corp., 850 Frelinghuysen Ave., Newark, and (for mail), Essex Fells, N. J.
CARRIER, Willis H.* (*M* 1913), (*Presidential Member*), (Pres., 1931; 1st Vice-Pres., 1930; 2nd Vice-Pres., 1929; Council, 1923-32), Chairman of the Bd. (for mail), Carrier Corp., 850 Frelinghuysen Ave., Newark, and Rensselaer Rd., Essex Fells, N. J.
CARROLL, William J. (*A* 1925), 1344 Broadway, Detroit, Mich.
CARSON, Clifford C. (*M* 1930), 15722 Detroit Ave., Cleveland, Ohio.
CARSTEN, William H. (*M* 1923), Majestic Furnace & Mfg. Co., 1723 Westlake Ave. N., Seattle, Wash.
CARTIER, C. Ernest, Jr. (*M* 1930), Associate, Htg. and Vtg. (for mail), Clinton S. Robison and Associates, Room 1812, 208 W. Washington St., and 7416 Rogers Ave., Chicago, Ill.
CASE, Walter G. (*A* 1930), Tech. Mgr., National Radiator Co., Ltd., Ideal House Gt., Marlborough St., London WI, and (for mail), 66 The Ridgeway- Kenton, Middlesex, England.
CASEY, Bryon L. (*M* 1921), Sales Engr. (for mail), Ilg Electric Vtg. Co., 182 N. LaSalle St., Chicago, and 423 N. Prospect Ave., Park Ridge, Ill.
CASEY, Huntley F. (*M* 1931), Mech. Engr., Construction Div., War Dept., Washington, D. C., and (for mail), P. O. Box 271, E. Falls Church, Va.
CASH, Tidie T. (*A* 1925), Mgr. (for mail), Grinnell Co., Inc., 240 Seventh Ave. S., and 617 Kenwood Pkwy., Minneapolis, Minn.
CASPERD, Henry W. H. (*J* 1930), Carrier Engrg. Co., Ltd., 12 Mission Row, Calcutta, India.
CASSELL, John D.* (*Life Member; M* 1913), (Council, 1930-33), 740 Garfield Ave., Palmyra, N. J.
CHALLMAN, Samuel A. (*M* 1919), Director of School Bldgs. for Minn. (for mail), State Dept. of Education, State Capitol, St. Paul, Minn.
CHANDLER, Clark W. (*S* 1930), (for mail), Chandler Co., and 1815 Ridgewood Ter., Cedar Rapids, Iowa.
CHAPIN, C. Graham (*M* 1933), 231 State St., New London, Conn.
CHAPPELL, Henry D. (*M* 1931), Dist. Mgr., V-Belt Drive Co., 100 Morgan Bldg., Detroit, and (for mail), 68 S. Johnson Ave., Pontiac, Mich.
CHARLTON, John Felder (*A* 1932), Engr. Appraiser (for mail), Box 2087, and 1631 N.E., Fifth St., Ft. Lauderdale, Fla.
CHASE, Chauncey L. (*M* 1931), Associate, Kopf & Sears, Consulting Engrs., 13 East 37th St., New York, and (for mail), 8829 Ft. Hamilton Pkwy., Brooklyn, N. Y.
CHASE, Louis R. (*J* 1931), Br. Mgr. (for mail), Buffalo Forge Co., 217 Dwight Bldg., and 4822 Wornall Rd., Kansas City, Mo.
CHERNE, Realto E. (*J* 1929), Engr. (for mail), 916 Temple Bldg., and 280 Westfield St., Rochester, N. Y.
CHERRY, Lester A.* (*M* 1921), Construction Engr. (for mail), Industrial Planning Corp., 43 Court St., Buffalo, and 155 Euclid Ave., Kenmore, Erie Co., N. Y.
CHERVEN, Victor W. (*M* 1928; *A* 1920), Chief Engr. (for mail), Holland Furnace Co., and 326 Maple Ave., Holland, Mich.
CHESTER, Thomas* (*M* 1917), Consulting Engr., 949 Chicago Blvd., Detroit, Mich.
CHEYNEY, Charles C. (*A* 1913), Buffalo Forge Co., 490 Broadway, Buffalo, N. Y.
CHOFFIN, C. C. (*M* 1919), Pres-Treas. (for mail), The W. J. Scholl Co., Mahoning Ave. and Hogue St., and 450 Catalina Ave., Youngstown, Ohio.
CHRISTENSON, Harry (*A* 1931), Supt. of Htg. (for mail), Hunter Prell Co., 311 Elm St., and 85 Wentworth Ave., Battle Creek, Mich.
CHRISTIAN, Charles W. (*M* 1913), Mgr. (for mail), Chas. W. Christian Co., P. O. Box 292, Charlotte, and 1101 Providence Rd., Myers Park, N. C.
CHRISTIE, Alfred Y. (*A* 1933), Salesman, U. S. Radiator Corp., 233 Vassar St., Cambridge, and (for mail), 715 LaGrange St., West Roxbury, Mass.
CHRISTMAN, William F. (*A* 1932; *J* 1931), Engr. (for mail), Kroeschell Engrg. Co., 2306 N. Knox Ave., and 3912 N. Hoyne Ave., Chicago, Ill.
CHURCH, Herbert John (*M* 1922), Mgr. (for mail), Darling Bros., Ltd., 137 Wellington St. W., Room 902, Toronto, and 358 Main St. N., Weston, Ont., Canada.
CLARE, Fulton Warren (*M* 1927), 141 Spring St. N.W., Atlanta, Ga.
CLARKE, Samuel S. (*Life Member; M* 1909), Pres. and Mgr. (for mail), S. S. Clarke & Co., Ltd., 605 Second St. W., and 603 Second St. W., Calgary, Alta, Canada.
CLARKSON, Robert C., Jr. (*M* 1921), 6050 Overbrook Ave., Philadelphia, Pa.
CLARKSON, W. B. (*M* 1919), 251 Broadway, Owatonna, Minn.
CLEGG, Carl (*M* 1922), Dist. Mgr. (for mail), American Blower Corp., 311 Mutual Bldg., and 3321 Gillham Rd., Kansas City, Mo.
CLEGG, Robert R. (*A* 1933), Zone Repr., Owens, Illinois Glass Co., Industrial Materials Div., Landreth Bldg., and (for mail), 4515 Lindell Blvd., St. Louis, Mo.
CLODFELTER, John L. (*A* 1932), Supt. (for mail), Carolina Sheet Metal Corp., 4210 Sansom St., Philadelphia, and West Chester Pike and Brief Ave., Elizabeth Manor Apt., Upper Darby, Pa.
CLOSE, Paul D.* (*M* 1928), Chief Engr., Industrial Uses Div. (for mail), Celotex Co., 919 N. Michigan Ave., Chicago, and 4622 Grove Niles Center, Ill.
CLOUGH, Leslie (*M* 1922), Consulting Engr. (for mail), Box 34, and 203 Pierce Rd., Weymouth, Mass.
COE, Ralph T. (*M* 1917), Prop. (for mail), The R. T. Coe Cos., 400 Reynolds Arcade, and 235 Chili Ave., Rochester, N. Y.
COHAGEN, Chandler C. (*M* 1919), P. O. Box 1305, Billings, Mont.
COHEN, Nathan (*S* 1933), 2305 Loring Pl., New York, N. Y.
COHEN, Philip (*M* 1932), Acting Dist. Mgr. (for mail), B. F. Sturtevant Co., 407 E. Ohio Gas Bldg., Cleveland, and 18519 Kinsman Blvd., Shaker Heights, Ohio.
COLBY, Clyde W. (*M* 1915), 2341 Carnegie Ave., Cleveland, Ohio.
COLCLOUGH, O. T. (*A* 1933), Custodian, American Legation, American Government Bldg., and (for mail), 407 Elgin St., Ottawa, Canada.

COLE, Edwin Q. (*M* 1931), 382 Lebanon St., Melrose, Mass.
COLE, Grant E. (*A* 1925), 439 King St. W., Toronto, Ont., Canada.
COLEMAN, John B. (*M* 1920), Chief Engr. (for mail), Grinnell Co., Inc., 275 W. Exchange St., and 237 Cole Ave., Providence, R. I.
COLLAMORE, Ralph (*M* 1904), (Bd. of Governors, 1913), Secy., Smith, Hinchman & Grylls, 800 Marquette Bldg., and (for mail), 679 Pingree Ave., Detroit, Mich.
COLLIER, William I. (*M* 1921), W. I. Collier & Co., 522 Park Ave., Baltimore, Md.
COLLINS, John F. S., Jr. (*M* 1933), Supervisor of Steam Installations (for mail), Allegheny County Steam Htg. Co., Philadelphia Co. Bldg., 435 Sixth Ave., and 827 N. Euclid Ave., Pittsburgh, Pa.
COMSTOCK, Glen Moore (*A* 1926), Dist. Repr. (for mail), L. J. Wing Mfg. Co., 154 College Ave., Beaver, Pa.
CONNELL, Richard F. (*M* 1916), Mgr. Capitol Testing Lab., U. S. Radiator Corp., 1056 First Natl. Bk. Bldg., Detroit, Mich.
CONNER, Raymond M. (*M* 1931), Director, A. G. A. Lab., Gas Engrg., and (for mail), 271 East 216th St., Cleveland, Ohio.
COOK, Benjamin F. (*M* 1920), Consulting Engr. (for mail), 114 W. Tenth St., Bldg., Kansas City, and 1720 Overton Ave., Independence, Mo.
COOK, Chauncey J. (*A* 1930), 121 Lyndhurst St., Syracuse, N. Y.
COOK, Howard A. (*A* 1933), Supt., University Plbg. & Htg. Co., 3939 University Way, and (for mail), 1433–33rd Ave., Seattle, Wash.
COOK, Ralph P. (*M* 1930), Engr. of Mech. Equip. (for mail), Eastman Kodak Co., Kodak Park, and 105 Falleson Rd., Rochester, N. Y.
COOMBE, James (*A* 1932), Vice-Pres. (for mail), The Wm. Powell Valve Co., 2525 Spring Grove Ave., and 2363 Grandeu Rd., Cincinnati, Ohio.
COON, Thurlow E. (*M* 1916), Pres. (for mail), The Coon-De Visser Co., 2051 W. Lafayette, and 826 Edison Ave., Detroit, Mich.
COOPER, Frederick D. (*A* 1930), Sales Engr., 905 Holden Ave., and (for mail), 1746 Longfellow Ave., Detroit, Mich.
COOPER, John R. (*M* 1931), East. Br. Mgr. (for mail), Thermal Units Mfg. Co., 30 Church St., and 680 Madison Ave., New York, N. Y.
COOPER, John W. (*M* 1932; *A* 1925; *J* 1921), Buffalo Forge Co., 1596 Arcade Bldg., St. Louis, Mo.
COPPERUD, Edmund R. (*J* 1933), Minneapolis Plbg. Co., 1420 Nicollet Ave., Minneapolis, Minn.
CORBIN, William E. (*J* 1929), 158 Laurel St., Buffalo, N. Y.
CORLETT, Lawrence D. (*M* 1930), 401 N. Emerson St., Detroit, Mich.
CORNELL, J. Clarence (*A* 1930), Checker (Mechanical) 12 South 12th St., and (for mail), 2823 W. Allegheny Ave., Philadelphia, Pa.
CORNWALL, George I. (*M* 1919), Mgr., Boiler Dept. (for mail), Hitchings & Co., 701 Spring St., and 633 Madison Ave., Elizabeth, N. J.
CORRAO, Joseph (*J* 1933), Estimator and Engr., Herman Lawson Co., 465 Tehama St., and (for mail), 854–31st Ave., San Francisco, Calif.
CORRIGAN, James A. (*S* 1930), 2501 W. St. Louis Ave., St. Louis, Mo.
COTTER, Leonard F. (*J* 1929), 34 Pearl St., Springfield, Mass.
COUGHLIN, Robert J. (*M* 1925), 1526 Oakwood Blvd., Royal Oak, Mich.
COWARD, Herbert (*M* 1921), Carrier Engrg. Corp., 604 Washington Bldg., Washington, D. C.
COX, Harrison F. (*A* 1930), 243 Carroll St., Paterson, N. J.
COX, William W. (*M* 1923), (for mail), Heating Service Co., 326 Columbia St., and 6232 31st Ave. N.E., Seattle, Wash.
CRANSTON, William E., Jr. (*M* 1931), Vice-Pres. (for mail), Thermador Electrical Mfg. Co., 116 Llewellyn St., Los Angeles, and 1912 Meridian Ave., South Pasadena, Calif.
CRESSY, Ralph E. (*J* 1929), Sales Engr., Hoffman Specialty Co., Chrysler Bldg., New York, and (for mail), 408 St. Lawrence Ave., Buffalo, N. Y.
CRIQUI, Albert A.* (*M* 1919), Chief Engr., Htg. and Vtg. Dept., Buffalo Forge Co., 490 Broadway, and (for mail), 250 Blaine Ave., Buffalo, N. Y.
CROFT, Terrell (*M* 1924), Apartado 563, Mexico City, Mexico.
CRONE, Charles E., Jr. (*M* 1922), Secy-Treas. (for mail), Wendt & Crone Co., 2124 Southport Ave., and 1320 N. State St., Chicago, Ill.
CRONE, Thomas E. (*M* 1920), 601 West 110th St., Apt. 10 M., New York, N. Y.
CROSS, Robert E. (*A* 1931), 95 State St., Springfield, Mass.
CUCCI, Victor J. (*M* 1930), Consulting Engr., Cucci & Brown, Inc., 347 Madison Ave., New York, N. Y.
CULBERT, William P. (*A* 1929), Secy. (for mail), Culbert-Whitby Co., Inc., 2019 Rittenhouse St., Philadelphia, and 929 Alexander Ave., Drexel Hill, Pa.
CUMMING, Robert W. (*M* 1928), Mech. and Sales Engr., Sarco Co., Inc., 183 Madison Ave., New York, and (for mail), 81 Alkamont Ave., Scarsdale, N. Y.
CUMMINGS, Carl H. (*A* 1927; *J* 1926), Mgr. (for mail), Industrial Appliance Co. of New England, 250 Stuart St., Boston, and 41 Edgehill Rd., Chestnut Hill, Mass.
CUMMINGS, Charles A. (*M* 1929; *A* 1929; *J* 1926), 38A Moreland St., Winter Hill, Mass.
CUMMINGS, G. J. (*M* 1923), 2001 Hoover Ave., Oakland, Calif.
CUMMINS, George H. (*M* 1919), Dist. Mgr. (for mail), Aerofin Corp., United Artists Bldg., and 17376 Wisconsin Ave., Detroit, Mich.
CUNNINGHAM, Thomas M. (*M* 1931; *A* 1931; *J* 1930), Vice-Pres. (for mail), Carrier Engrg. Corp. of Texas, 2022 Bryan St., and 5712 Marquita St., Dallas, Texas.
CURRIER, Charles H. (*M* 1919), Pres. (for mail), Drying Systems, Inc., 1800 Foster Ave., and 2440 Lakeview Ave., Chicago, Ill.
CUSHMAN, Lester D. (*M* 1930), 89 Traincroft St., Medford, Mass.
CUTLER, Joseph A. (*M* 1916), (Council, 1917-1926), Vice-Pres. (for mail), Johnson Service Co., 1355 Washington Blvd., Chicago, and 649 Hinman Ave., Evanston, Ill.

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DALLA VALLE, J. M.* (*J* 1933), Asst. Sanitary Engr. (for mail), U. S. Public Health Service, 19th and Constitution Ave., Washington, D. C., and 6409 Georgia St., Chevy Chase, Md.
DALY, John H. (*M* 1915), Mfrs. Agent, 1544 Welton St., Denver, Colo.
DALY, Robert E. (*M* 1931), Executive Dept. (for mail), American Radiator Co., 40 West 40th St., and 12 East 88th St., New York, N. Y.
DAMBLY, A. Ernest (*M* 1924; *J* 1921), (for mail), H. B. Hackett, 901 Architects Bldg., Philadelphia, Pa., and Harvey Cedars, N. J.
DANE, Irving S. (*M* 1925), 166 George St., Medford, Mass.
DANFORTH, N. Loring (*M* 1919), John W. Danforth Co., 72 Ellicott St., Buffalo, N. Y.
DARLING, Arthur B. (*A* 1929), Asst. Sales Mgr. (for mail), Darling Bros., Ltd., 140 Prince St., and 1935 St. Luke St., Montreal, P. Q., Canada.
DARLINGTON, Allan P. (*M* 1930), Salesman (for mail), American Blower Corp., 2539 Woodward Ave., and 3605 Devonshire Rd., Detroit, Mich.
DARTS, John A. (*M* 1919), Kewanee Boiler Co., Inc., 570 Seventh Ave., New York, N. Y.
DAUCH, Emil O. (*M* 1921), Secy-Treas. (for mail), McCormick Plbg. Supply Co., 1675 Bagley Ave., and The Wilshire Hotel, Detroit, Mich.

DAVENPORT, R. F. (*A* 1933), 77 James St. E., Brockville, Ont., Canada.
DAVIDSON, L. Clifford (*M* 1927), Associate Dist. Mgr. (for mail), Buffalo Forge Co., 220 South 16th St., and 6312 Sherwood Rd., Philadelphia, Pa.
DAVIDSON, Phillip L. (*M* 1924; *J* 1921), Carrier Engrg. Corp., 12 South 12th St., Philadelphia, Pa.
DAVIES, George William (*M* 1918), Managing Director (for mail), Htg., Vtg. and Domestic Engrs., 79 Maclaggan St., Dunedin, and 145 Kenmure Rd., Mornington, New Zealand.
DAVIS, Arthur C.* (*M* 1920), Asst. Supt., The Holland Tunnel, Canal and Varick Sts., New York, N. Y., and (for mail), 73 Preston St., Ridgefield Park, N. J.
DAVIS, Bert C. (*M* 1904), (Council, 1917), Pres-Treas. (for mail), American Warming & Vtg. Co., 317-19 Pennsylvania Ave., and 603 W. Church St., Elmira, N. Y.
DAVIS, Calvin R. (*M* 1927), Br. Mgr. (for mail), Johnson Service Co., 2328 Locust St., St. Louis, and 7534 Westmoreland Pl., Clayton, Mo.
DAVIS, Herbert H. (*A* 1929), Herbert H. Davis Co., Inc., 4146 S. Western Ave., Chicago, Ill.
DAVIS, Joseph (*M* 1927; *A* 1926), Htg. and Vtg. Contractor, 607 Root Bldg., Buffalo, and (for mail), 85 Warren Ave., Kenmore, N. Y.
DAVIS, Leo J. (*M* 1917), 18261 Grayfield Ave., Detroit, Mich.
DAVIS, Otis E. (*M* 1929; *A* 1925), 1501 Fourth Ave., Scotts Bluff, Nebr.
DAVIS, Robert J. (*A* 1933), Supt. Construction (for mail), Metropolitan Bldg. Co., 1201 Fourth Ave., and 6318–39th Ave. S.W., Seattle, Wash.
DAVIS, Rowland G. (*A* 1921), Sales Engr., Herman Nelson Corp., 400 Ninth Vincent Bldg., Cleveland, and (for mail), 887 Nela View Rd., Cleveland Heights, Ohio.
DAWSON, Thomas L. (*M* 1930), Pres. (for mail), Thomas L. Dawson Co., 2035 Washington St., Kansas City, Mo., and 56th and Shawnee Mission Rd., Rosedale Sta., Kansas City, Kans.
DAY, V. S.* (*M* 1924), Engr. (for mail), Carrier Corp., 850 Frelinghuysen Ave., Newark, and 18 Roosevelt Rd., Summit, N. J.
DEAN, Charles L. (*M* 1932), Asst. Prof. Mech. Engrg., University of Wisconsin, and (for mail), 2603 Stevens St., Madison, Wis.
DEELY, James J. (*J* 1933), House Htg. (Resident) Sales Engr., Brooklyn Union Gas Co., 180 Remsen St., Brooklyn, and (for mail), 61 College Ave., Tarrytown, N. Y.
DEEX, Charles J. (*M* 1920), Mouat-Vapor Htg. Co., 1246 W. Fourth St., Cleveland, Ohio.
DeLANCEY, Ralph W. (*A* 1932; *J* 1930), 1000 W. Fifth St., Winona, Minn.
DeLAND, Charles W. (*M* 1924; *J* 1923), Secy-Treas. (for mail), C. W. Johnson Co., Inc., 211 N. Desplaines St., and 2021 Estes Ave., Chicago, Ill.
DEMPSEY, Harry P. (*M* 1919), 394 Pleasant Ave., Hamburg, N. Y.
DEUTCHMAN, Julius (*S* 1933), 1-3 Wellesley Ave., Yonkers, N. Y.
DIBBLE, S. E.* (*M* 1917), (*Presidential Member*), (Pres., 1925; 1st Vice-Pres., 1924; 2nd Vice-Pres., 1922; Council, 1921-1926) Supt., Patton School, Elizabethtown, Pa.
DICE, Eugene S. (*S* 1933), 7141 Upland St., Pittsburgh, Pa.
DICKEY, Arthur J. (*M* 1921), Vice-Pres.-Gen. Mgr. (for mail), C. A. Dunham Co., Ltd., 1523 Davenport Rd., and 9 Mosson Pl., Toronto, Ont., Canada.
DICKSON, Robert B. (*M* 1919), Pres. (for mail), Kewanee Boiler Corp., Franklin St and Q Tracks, and 409 E. Prospect St., Kewanee, Ill.
DIGBY, Homer E. (*A* 1925; *J* 1922), 216 Oneida St., Pittsburgh, Pa.
D'IMOR, Elton J. (*M* 1933), Mgr. and Engr., The Trane Co., LaCrosse, Wis., and (for mail), 44 S. Tucker, Memphis, Tenn.
DISTEL, Frank (*M* 1918), P. O. Box 133, Lansing, Mich.
DIVER, M. L. (*M* 1925), Consulting Engr., P. O. Box 1016, San Antonio, Texas.
DIXON, Arthur G. (*M* 1928), Sales Mgr. (for mail), Modine Mfg. Co., and 442 Wolff St., Racine, Wis.
DOBBS, C. E. (*A* 1921), Repr., Burnham Boiler Corp., 31st and Jefferson Sts., Philadelphia, Pa., and (for mail), 72 Berlin Ave., Haddonfield, N. J.
DOCK, Chester J. (*M* 1931), 132 S. Tenth St., Minneapolis, Minn.
DOCKERAY, Fayette (*A* 1931), Mfrs. Agent, Htg. and Vtg. Specialty, Box 135, Kingston, Pa.
DODDS, Forrest F. (*M* 1920), Br. Mgr. (for mail), American Radiator Co., 1021 Grand Ave., and 235 Ward Pkwy., Kansas City, Mo.
DOERING, Frank L. (*M* 1919), Salesman, American Radiator Co., 219 Denver Ave., Lynchburg, Va.
DOHERTY, John J. (*M* 1921), P. C. Doherty Co., 114 Main St., Poughkeepsie, N. Y.
DOHERTY, Russell (*A* 1920), Asst. Chicago Mgr. (for mail), National Radiator Corp., 1111 East 83rd St., Chicago, and 300 Forest Ave., Oak Park, Ill.
DOLAN, Hugh P. (*M* 1930), 14419 Strathmoor Ave., Detroit, Mich.
DOLAN, Raymond G. (*M* 1926; *A* 1926; *J* 1922), Secy-Treas. (for mail), Tom Dolan Htg. Co., Inc., 614 W. Grand, and 2112 West 20., Oklahoma City, Okla.
DONNELLY, James A.* (*M* 1904), (Treas., 1912-1914), Largent, W. Va.
DONNELLY, Russell (*M* 1923), Sales Engr. (for mail), Nash Engrg. Co., Graybar Bldg., 420 Lexington Ave., New York, N. Y.
DONOVAN, William J. (*A* 1930), 2230 North 27th St., Philadelphia, Pa.
DONZELLI, Enrico (*M* 1933), Piazza SS Pietro e Lino No. 4, Milan, Italy.
DORFAN, M. I. (*M* 1929), Mgr. Dust Collecting Div., Blaw-Knox Co., P. O. Box 915, and (for mail), 6436 Nicholson St., Pittsburgh, Pa.
DORNHEIM, G. A. (*M* 1912; *J* 1906), 15 Hamilton Ave., Bronxville, N. Y.
DORSEY, Francis C. (*M* 1920), Engr. and Contractor (for mail), Francis C. Dorsey, Inc., 4520 Schenley Rd., Roland Park, and 212 Gittings Ave., Baltimore, Md.
DOUGHTY, Charles John (*M* 1925), Pres. and Managing Director (for mail), C. J. Doughty & Co., Fed., Inc., U. S. A., 30 Brenan Rd., and 1920 Ave Joffre, Shanghai, China.
DOWNE, Edward R. (*M* 1927), American Gas Products Corp., 40 West 40th St., New York, N. Y.
DOWNE, Henry S. (*Life Member; M* 1895), Cie Nationale des Radiateurs, 149 Boulevard Haussman, Paris, France.
DOWNS, Nate W. (*M* 1917), (Council, 1928-1930), Chief Engr. and Supt. of Bldgs. (for mail), School Dist. of Kansas City, Mo., 317 Finance Bldg., and 2119 East 68th St., Kansas City, Mo.
DOWNS, Sewell H. (*M* 1931), Chief Engr., Clarage Fan Co., and (for mail), 211 Creston Ave., Kalamazoo, Mich.
DOYLE, William J. (*M* 1920), Factory Mgr., The Williamson Heater Co., 4558 Marburg Ave., and (for mail), 3766 Hyde Park Ave., Cincinnati, Ohio.
DRESEN, William D. (*M* 1929; *A* 1920; *J* 1926), 3506–73rd St., Jackson Heights, L. I., N. Y.
DRINKER, Philip* (*M* 1922), Assoc. Prof. (for mail), Harvard School of Public Health, 55 Van Dyke St., Boston, and 11 Lowell Rd., Brookline, Mass.
DRISCOLL, William H.* (*M* 1904), (*Presidential Member*), (Pres., 1926; 1st Vice-Pres, 1925; 2nd Vice-Pres., 1924; Treas., 1923; Council, 1918-1927), Vice-Pres. (for mail), Thompson-Starrett Co., Inc., 250 Park Ave., New York, N. Y., and 65 Union St., Montclair, N. J.
DUBÉ, Wilbrod (*M* 1925), 316 Laurier Ave., Quebec, Canada.
DuBOIS, Louis J. (*M* 1931), Air Cond. Engr., York Ice Machinery Corp., 117 South 11th St., and (for mail), 7337a Lindell Ave., St. Louis, Mo.

DUBRY, Ernest E. (*M* 1924), Asst. Supt., Central Htg., The Detroit Edison Co., 2000 Second Ave., and (for mail), 9116 Dexter Blvd., Detroit, Mich.
DUDLEY, William Lyle (*M* 1922), Western Blower Co., 1800 Ninth Ave. S., Seattle, Wash.
DUFF, Kennedy (*M* 1915), Mgr. (for mail), Johnson Service Co., 28 East 29th St., New York, and 9 Park Ave., Maplewood, N. J.
DUFFIELD, Thomas J.* (*A* 1927), 525 West 238th St., Apt. 4-L, New York, N. Y.
DUGAN, Thomas M. (*M* 1920), Sanitary and Htg. Engr., Natl. Tube Co., Fourth Ave. and Locust St., and (for mail), 1308 Freemont St., McKeesport, Pa.
DUNBRACK, Alvin P. (*J* 1933), 172 Arlington Ave., Jersey City, N. J.
DUNCAN, George W., Jr. (*M* 1923), 2512 Benvenue Ave., Berkeley, Calif.
DUNCAN, James R. (*M* 1923), Carrier Australasia, Ltd., 56 Hunter St., Sydney, Australia.
DUNCAN, William A. (*A* 1930), Dist. Service Engr. (for mail), Dominion Oxygen Co., Ltd., 92 Adelaide St. W., and 20 Tyrell Ave., Toronto, Ont., Canada.
DUNHAM, Clayton A.* (*M* 1911), Pres. (for mail), C. A. Dunham Co., 450 E. Ohio St., Chicago, and 150 Maple Hill Rd., Glencoe, Ill.
DURKEE, Merritt E. (*A* 1930), Sales Engr. (for mail), C. A. Dunham Co., 101 Park Ave., New York, and 254 Martine Ave., White Plains, N. Y.
DURNING, Edward H. (*J* 1931), Commercial Sales, Community Natural Gas Co., 1915 Wood St., and (for mail), 4937 Worth St., Dallas, Texas.
DURYEA, Albert A. (*S* 1933), Belden Point, City Island, N. Y.
DUSOSSOIT, Edmond A. (*M* 1920), Treas. (for mail), Lynch & Woodward, Inc., 320 Dover St., Boston, and 16 Hancock Ave., Newton Centre, Mass.
DWYER, Thomas F. (*M* 1923), Mech. Engr. (for mail), Board of Education, 49 Flatbush Ave. Ext., Brooklyn, and 1242 Morris Ave., New York, N.Y.
DYER, Orville K. (*M* 1919), Mgr. Blower Div. (for mail), Buffalo Forge Co., 490 Broadway, and 11 Russell Ave., Buffalo, N. Y.

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EADIE, John G. (*M* 1909), Eadie, Freund & Campbell Co., 110 West 40th St., New York, N. Y.
EAGAN, Walter H. (*M* 1926), Walter H. Eagan & Co., 1612 Vine St., Philadelphia, Pa.
EAGAR, R. Frank (*M* 1922), 89 Hollis St., Halifax, Nova Scotia.
EAKINS, Walter (*M* 1928), Vice-Pres. (for mail), Louis J. Sommer & Son, Inc., 2436 Brown St., Philadelphia, and 336 E. Phil Ellena St., Germantown, Philadelphia, Pa.
EASTERBROOKS, Clifton C. (*M* 1922), Sales Engr. (for mail), Koithan & Pryor, 39 Cortlandt St., New York, N. Y., and 722 Mountain Ave., Westfield, N. J.
EASTMAN, Carl B. (*M* 1932; *A* 1932; *J* 1929), Mgr. Philadelphia, Sales Office., C. A. Dunham Co., 1500 Walnut St., Philadelphia, and (for mail), 7247 Calvin Rd., Upper Darby, Pa.
EASTWOOD, E. O. (*M* 1921), (Council, 1931-1933), Prof. of Mech. Engrg. (for mail), University of Washington, and 4702 12th Ave. N.E., Seattle, Wash.
EATON, Byron K. (*M* 1920), Regional Sales Mgr., Delco Appliance Corp., and (for mail), 2018 Binney St., Omaha, Nebr.
EATON, Vincent (*M* 1930), Consulting Engr., 3030 Euclid Ave., Cleveland, and (for mail), 15702 Hazel Rd., E. Eleveland, Ohio.
EBERT, William A. (*M* 1920), 1004 Drexel Ave., San Antonio, Texas.
EDWARDS, Daniel F. (*M* 1920), 2340-42 Pine St., St. Louis, Mo.
EDWARDS, Don C., Jr. (*J* 1931), London, Ky.
EDWARDS, Don J. (*A* 1933), Vice-Pres. (for mail), General Heat & Appliance Co., 94 Massachusetts Ave.. Boston. and 40 Rockledge Rd., Newton, Mass
EDWARDS, Paul A. (*M* 1919), Pres. (for mail), The G. F. Higgins Co., 608 Wabash Bldg., and 3074 Pinehurst Ave., Pittsburgh, Pa.
EELLS, Henry B. (*M* 1926), New York Mgr., Barnes & Jones, Inc., 101 Park Ave., New York, and (for mail), 1049 East 27th St., Brooklyn, N. Y.
EGGLESTON, Lewis W. (*M* 1921), American Radiator Co., 5961 Lincoln Ave., Detroit, Mich.
EGGLY, Harry J, Jr. (*M* 1933), Consulting Engr. (for mail), 1805 Walnut St., Philadelphia, and Elkins Park Apts., Elkins Park, Pa.
EHRLICH, M. William* (*M* 1916), Chief Engr., Commodore Heaters Corp., 11 West 42nd St., New York, N. Y., and (for mail), 56 Ridge Rd., Lyndhurst, N. J.
EICHBERG, W. Roy (*M* 1929), Pres. (for mail), Carolina Sheet Metal Corp., 4210 Sansom St., Philadelphia, and 828 Turner Ave., Drexel Hill, Pa.
EICHER, HuBert C. (*M* 1922), State Director, School Bldgs. Div., Dept. of Public Instruction, State Capitol, and (for mail), 103 South St., Harrisburg, Pa.
EISS, Robert M. (*M* 1933; *A* 1933; *J* 1930), (for mail), 72 Brantwood Rd., c/o Buffalo N.Y.P.O., Eggertsville, N. Y.
ELLINGWOOD, Elliott L. (*M* 1909), Consulting Engr. (for mail), 354 S. Spring St., Los Angeles, Calif.
ELLIOT, Edwin (*M* 1929), (for mail), Edwin Elliot & Co., 560 North 16th St., Philadelphia, and 403 W. Price St., Germantown, Philadelphia, Pa.
ELLIOTT, Louis (*M* 1932), Consulting Mech. Engr., Electric Bond & Share Co., 2 Rector St., Room 1914, New York, N. Y.
ELLIS, Ernest E. (*M* 1922), Secy-Treas., F. A. Ellis & Co., Inc., 840 Center St., Winnetka, Ill.
ELLIS, Frederick E. (*M* 1923), Sales Mgr. (for mail), Imperial Iron Corp., Ltd., 30 Jefferson Ave., and 372 Durie St., Toronto, Ont., Canada.
ELLIS, Frederic R. (*M* 1913), Sales Engr., Buerkel & Co., Inc., 18-24 Union Park St., Boston, and (for mail), 131 Beacon St., Hyde Park, Mass.
ELLIS, Harry W. (*M* 1923; *A* 1909), Pres., Johnson Service Co., 507 E. Michigan St., Milwaukee, Wis.
EMERSON, Ralph R. (*M* 1922), 48 Gay St., Newtonville, Mass.
EMERY, Hugh (*S* 1933), 20 Morgan Pl., N. Arlington, N. J.
EMMERT, Luther D. (*M* 1919), Repr. (for mail), Buffalo Forge Co., 15 N. Jefferson St., Chicago, and 1704 Hinman Ave., Evanston, Ill.
ENGEL, Edward (*J* 1933), Design Draftsman (for mail), U. S. Navy Yard, Hull Div., and 2608 North 30th St., Philadelphia, Pa.
ENGLE, Alfred (*A* 1923), Sales Mgr. (for mail), Jenkins Bros., 80 White St., New York, and 1 Edgewood Rd., Scarsdale, N. Y.
ERDLE, Gardner F. (*A* 1933), (for mail), 374 Delaware Ave., Buffalo, and 19 Tremont Ave., Kenmore, N. Y.
ERICKSON, Harry H. (*A* 1929), Sales Engr., National Radiator Co., and (for mail), 5909 North 21st St., Philadelphia, Pa.
ERICKSON, Martin E. (*A* 1926), Supt., Maintenance, Bd. of Education, and (for mail), 1533 South 74th St., West Allis, Wis.
ERICSSON, Eric B. (*M* 1933), Engr., Custodian, Bd. of Education, and (for mail), 605 West 116th St., Chicago, Ill.
ETHERIDGE, G. Thomas, Jr. (*S* 1931), 118 Main St., Belle Vernon, Pa.
EVANS, C. A. (*M* 1919), 218 Lexington Ave., Buffalo, N. Y.
EVANS, Edwin C. (*M* 1919), 2953 Zephyr Ave., Corliss Sta. P. O., Pittsburgh, Pa.
EVANS, William A. (*M* 1918), 24 Woodland Rd., Maplewood, N. J.
EVELETH, Charles F.* (*M* 1911), 2030 East 115th St., Cleveland, Ohio.
EVERETTS, John, Jr. (*J* 1929), Engr. (for mail), W. L. Fleisher, 11 West 42nd St., New York, and 83 Kenilworth Pl., Brooklyn, N. Y.

EVLETH, Everett B. (*A* 1927), Br. Mgr. (for mail), Minneapolis-Honeywell Regulator Co., 2831 Olive St., and 1138 Boland Pl., St. Louis, Mo.
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FAHNESTOCK, Maurice K.* (*M* 1927), Special Research Asst., Prof. (for mail), University of Illinois, 214 M. E. Lab., and 701 W. California St., Urbana, Ill.
FALTENBACHER, Harry J. (*M* 1930), 235 E. Wister St., Philadelphia, Pa.
FALVEY, John D. (*M* 1922), Hester-Bradley Co., 2835 Washington Ave., St. Louis, Mo.
FAMILETTI, A. Robert (*J* 1930), 2230 Tasker St., Philadelphia, Pa.
FANSLER, P. E. (*A* 1927), Editor, Oil Heat (for mail), 167 Madison Ave., New York, N. Y., and Catonsville, Md.
FARLEY, John W. (*A* 1921), Mgr., Farley Sleeve & Hanger Co., 3748 East 71st, Cleveland, Ohio.
FARLEY, W. F. (*M* 1930), Salesman, American Radiator Co., 40 West 40th St., New York, and (for mail), 28 Elm St., New Rochelle, N. Y.
FARLEY, Willoughby S. (*J* 1933), Partner, Farley & Luther, 211 Montague St., Danville, Va.
FARNHAM, Roswell (*M* 1920), Council, 1927: 1933), Dist. Mgr., Engrg. Sales (for mail), Buffalo Forge Co., P. O. Box 985, and 5 Clarendon Pl., Buffalo, N. Y.
FARNSWORTH, John G. (*J* 1931), Gas House Htg. Engr. (for mail), Central Illinois Light Co., 316 S. Jefferson St., and 313 Crescent St., Peoria, Ill.
FARRAR, Cecil W. (*M* 1920; *A* 1918), (Treas., 1930; Council, 1930), Pres. (for mail), Excelso Products Corp., 1807 Elmwood Ave., and 29 Oakland Pl., Buffalo, N. Y.
FAUST, Frank H.* (*J* 1930), Engr. (for mail), General Electric Co., 1 River Rd., and 114 Union St., Schenectady, N. Y.
FAY, Francis C. (*M* 1925), Raisler Htg. Co., 129-31 Amsterdam Ave., New York, N. Y.
FEBREY, Ernest J. (*M* 1903), Htg. and Air Cond. (for mail), 616 New York Ave. N.W., and 1610 Riggs Pl. N.W., Washington, D. C.
FEEHAN, John B. (*M* 1923), Pres-Treas. (for mail), John B. Feehan, Inc., 471 Union St., Lynn, and 4 Ocean View Dr., Marblehead, Mass.
FEELY, Frank J. (*A* 1929), 17215 Greenlawn, Detroit, Mich.
FEGLEY, Donald R. (*S* 1933), 2274 Loring Pl., New York, N. Y.
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HATTON, Albert E. (*M* 1931), Supt. and Estimator (for mail), Lome Plbg. & Htg. Co., 641 W. Congress, Detroit, and R. R. No. 1, Birmingham, Mich.

HAUAN, Merlin J. (*M* 1933), Consulting Engr., 3412–16th S., Seattle, Wash.

HAUPT, Howard F. (*A* 1929), 614 E. Beaumont Ave., Milwaukee, Wis.

HAUSS, Charles F.* (*Charter Member; Life Member; M* 1922), 305 Southern Ave., Cincinnati, Ohio.

HAYDEN, Carl F. (*A* 1930), Br. Mgr. (for mail), Barber-Colman Co., 221 N. LaSalle St., Chicago, and 2309 Forestview Rd., Evanston, Ill.

HAYES, James J. (*M* 1920), Sales Engr. (for mail), Room 925, 53 W. Jackson Blvd., and 7443 Jeffrey Ave., Chicago, Ill.

HAYES, John J. (*A* 1933), Auburn Stoker Sales Corp., 406 N. Wells St., Chicago, and (for mail), 918 Michigan Ave., Evanston, Ill.

HAYES, Joseph G. (*M* 1908), Pres. and Engr. (for mail), Hayes Bros., Inc., 236 W. Vermont St., and 2849 N. Capitol Ave., Indianapolis, Ind.

HAYMAN, A. Eugene, Jr. (*S* 1930), 2500 Washington St., Wilmington, Del.

HAYNES, Charles V. (*M* 1917), (1st Vice-Pres., 1933; 2nd Vice-Pres., 1932; Council, 1926-1929, 1933), Vice-Pres., Hoffman Specialty Co., 3709 Chrysler Bldg., New York, N. Y., and (for mail), 115 Llanfair Rd., Ardmore, Mont. Co., Pa.

HAYWARD, Ralph B. (*M* 1909), Pres. (for mail), R. B. Hayward Co., 1714 Sheffield Ave., Chicago, and 201 S. Stone Ave., LaGrange, Ill.

HEAGLER, John M. (*M* 1922), 1646 Igelhart Ave., St. Paul, Minn.

HEARD, John A. E. (*J* 1930), Carrier Engrg. Co., Ltd., Sardar Sujan Singh Block, Connaught Pl., New Delhi, India.

HEARD, Roderick G. (*A* 1933), Oil Burner Dept. (for mail), Imperial Oil, Ltd., 56 Church St., and 6 Walker Ave., Toronto, Ont., Canada.

HEATH, Frederick R. (*M* 1913), Sales Engr. (for mail), E. B. Badger & Sons Co., 75 Pitts St., Boston, and 89 Trowbridge St., Cambridge, Mass.

HEATH, William R. (*M* 1931), 674 Crescent Ave., Buffalo, N. Y.
HECHT, Frank H. (*M* 1930), Sales Engr. (for mail), B. F. Sturtevant Co., 2635 Koppers Bldg., and 1467 Barnesdale St., Pittsburgh, Pa.
HECK, George L., Jr. (*A* 1921), Chief Engr., Garden City Fan Co., McCormick Bldg., and (for mail), 8340 Maryland Ave., Chicago, Ill.
HECKEL, E. P. (*M* 1918), Vice-Pres. and Mgr., Chicago, Dist. (for mail), Carrier Engr. Corp., 180 N. Michigan Ave., Chicago, and 314 Cuttriss Pl., Park Ridge, Ill.
HEDGES, H. Berkley (*M* 1919), 1021 Park Lane, Plainfield, N. J.
HEDLEY, Park S. (*M* 1923), Park S. Hedley Co., Curtiss Bldg., Delaware at Tupper, Buffalo, N.Y.
HEEBNER, Walter M. (*M* 1922), Sales Engr., Warren Webster & Co., 470 Fourth Ave., New York, N. Y., and (for mail), 282 Highwood Ave., Teaneck, N. J.
HEIBEL, Walter E. (*M* 1917), Dist. Mgr. (for mail), Aerofin Corp., 11 West 42nd St., New York, N. Y., and Old Greenwich, Conn.
HEIDENREICH, George (*M* 1928), 325-326 Bd. of Trade Bldg., Indianapolis, Ind.
HEILMAN, Russell H.* (*M* 1923), Senior Industrial Fellow (for mail), Mellon Institute, and 5637 Wilkins Ave., Pittsburgh, Pa.
HELBURN, I. B. (*M* 1929; *J* 1927), Junior Assoc. (for mail), Wyman Engrg., Chamber of Commerce Bldg., and 700 Chalfonte Pl., Apt. 17., Cincinnati, Ohio.
HELLSTROM, John (*A* 1929), Vice-Pres. (for mail), American Air Filter Co., 215 Central Ave., Louisville, and Anchorage, Ky.
HELSTROM, Herman G. (*M* 1928), Sales (for mail), Kewanee Boiler Corp., 708 Builders Exchange, and 4608 Arden Ave., Minneapolis, Minn.
HEMINGWAY, William S. (*M* 1906), Mesa Grande, Calif.
HENDRICKSON, John J. (*A* 1932), Prod. Engr. (for mail), Bryant Heater & Mfg. Co., 17825 St. Clair Ave., Cleveland, and 1475 Genessee Rd., South Euclid, Ohio.
HENION, Hudson D. (*A* 1923), Sales Mgr. (for mail), C. A. Dunham Co., Ltd., 1523 Davenport Rd., and 45 Ridge Dr., Toronto, Ont., Canada.
HENRY, Alexander S., Jr. (*M* 1930), 300 Central Park W., New York, N. Y.
HERENDEEN, Frederick W. (*M* 1920), Secy., The Institute of Boiler and Radiator Mfrs., 29 Seneca St., Geneva, N. Y.
HERLIHY, Jeremiah J. (*Life Member; M* 1914), Pres. (for mail), J. J. Herlihy, Inc., 810 W. Congress St., and 3634 N. Keeler Ave., Chicago, Ill.
HERMAN, J., Jr. (*A* 1932; *J* 1930), 1349 E. Vernon Ave., Los Angeles, Calif.
HERRICK, Daniel A. (*M* 1923), Gen. Mgr. (for mail), Julian d'Este Co., 6 Spice St. (Charlestown Dist.), Boston, and 27 Agassiz St., Cambridge, Mass.
HERRING, Edgar (*M* 1919), Chairman and Governing Director (for mail), J. Jeffreys & Co., Ltd., Barrons Pl., Waterloo Rd., London S.E., and "Kenia" Keswick Rd., Putney, London S.W., England.
HERRMAN, Harold C. (*S* 1932), Instructor, Milwaukee Vocational School, and (for mail), 4523 North 22nd St., Milwaukee, Wis.
HERSH, Franklin C. (*A* 1933; *J* 1930), Carrier Engrg. Corp., 2022 Bryan St., Dallas, Texas.
HERSHEY, Leon A. (*A* 1933), 316 Huntington Ave., Boston, Mass.
HERTY, Frank B. (*M* 1933), House Htg. Supervisor, Brooklyn Union Gas Co., 176 Remsen St., and (for mail), 50 East 18th St., Brooklyn, N. Y.
HERTZ, H. Porter (*M* 1924), Routledge & Hertz, 204 Exchange Natl. Bk., Hutchinson, Kans.
HERTZLER, John R. (*J* 1928), Air Cond. Sales Engr. (for mail), York Ice Machinery Corp., 42nd St. and Second Ave., Brooklyn, and 1 University Pl., New York, N. Y.
HESS, David K. (*S* 1932), Student, University of Wisconsin, Madison, Wis., Hess Warming & Vtg. Co., 1211-1227 S. Western Ave., Chicago, Ill., and (for mail), 627 Mendota Court, Madison, Wis.
HESTER, Thomas J. (*M* 1919), Vice-Pres. and Treas. (for mail), Hester-Bradley Co., 2835 Washington Blvd., and 67 Aberdeen Pl., St. Louis, Mo.
HEXAMER, Harry D. (*M* 1931), Sales Engr. (for mail), Excelso Products Corp., 65 Clyde Ave., and 163 E. Delavan Ave., Buffalo, N. Y.
HEYDON, Charles G. (*A* 1923), Mgr. Sales of Western Div., Wright-Austin Co., 315 W. Woodbridge St., and (for mail), 2681 Nebraska, Detroit, Mich.
HIBBS, Frank C. (*M* 1917), Htg. Engr., The H. B. Smith Co., 2209 Chestnut St., and (for mail), 846 North 65th St., Philadelphia, Pa.
HICKEY, Daniel W. (*A* 1931), 278 W. Fourth St., St. Paul, Minn.
HICKEY, James W. (*S* 1932), P. O. Box 245, Carnegie Institute of Technology, Pittsburgh, Pa.
HICKS, William W. (*A* 1929), Mgr., W. W. Hicks & Co., 567 Banning St., Winnipeg, Man., Canada.
HIERS, Charles R. (*M* 1929; *A* 1929; *J* 1927), Apt. 10-D, 23 N. Columbus Ave., Mt. Vernon, N. Y.
HIGGINS, Dan T. (*M* 1928), McCarthy & Co., 529 S. Cascade Ave., Colorado Springs, Colo.
HIGGINS, Thomas J. (*M* 1927; *A* 1927; *J* 1923), Vice-Pres. (for mail), J. O. Ross Engrg. Corp., 122 East 42nd St., New York, N. Y.
HILL, Dr. E. Vernon* (*M* 1914; *A* 1912), (*Presidential Member*), (Pres., 1920; 1st Vice-Pres., 1919; 2nd Vice-Pres., 1918; Council, 1915-1921), Pres. (for mail), E. Vernon Hill Co., 121 N. Clark St., and 1126 Farwell Ave., Chicago, Ill.
HILL, Fred M. (*M* 1930), 225 East Ave. 39, Los Angeles, Calif.
HILLIARD, Charles E. (*M* 1932; *A* 1932; *J* 1927), Htg. and Vtg. Engr. (for mail), E. C. Hilliard Co., 27 B St., South Boston, and 1301 Washington St., South Braintree, Mass.
HILLIARD, Forrest H. (*A* 1930), 2508 S. Flower St., Los Angeles, Calif.
HILLS, Arthur H. (*M* 1924), Mgr. (for mail), Sarco Canada, Ltd., 1002 Federal Bldg., and 1005 Shaw St., Toronto, Ont., Canada.
HINCHMAN, Ernest G. (*M* 1923), 14 Nancy Blvd., Merrick, L. I., N. Y.
HINKLE, Edwin C. (*M* 1911), Htg. and Vtg. Engr., Atlantic Htg. & Engrg. Co., 170 N. Franklin St., Hempstead, L. I., N. Y.
HINRICHSEN, Arthur F. (*M* 1928), Pres-Treas., A. F. Hinrichsen, Inc., 50 Church St., New York, N. Y.
HIRES, J. Edgar (*M* 1927), Pres., Hires-Castner & Harris, Inc., 123 S. Broad St., Philadelphia, and (for mail), 107 Linwood Ave., Ardmore, Pa.
HIRSCHMAN, William F. (*M* 1929), Pres. and Chief Engr., W. F. Hirschman Co., Inc., 220 Delaware Ave., Buffalo, and (for mail), 165 Le Brun Circle, Eggertsville, N. Y.
HIRST, James Noble (*M* 1930; *J* 1927), Engr. (for mail), Carrier Products Corp., 850 Frelinghuysen Ave., Newark, and 146 DeHart Pl., Elizabeth, N. J.
HITCHCOCK, Frederick P. (*M* 1917), Buckeye Blower Co., 213-214 Lathrop Bldg., Kansas City, Mo.
HITCHCOCK, Paul C. (*M* 1931), 4939 Girard Ave. S., Minneapolis, Minn.
HJERPE, Clarence A., Jr. (*J* 1931), 73 Arch St., New Britain, Conn.
HOBBS, J. Clarence (*M* 1920), 60 Wood St., Painesville, Ohio.
HOCHULI, Henry W. (*M* 1925), Sales Engr., Natl. Radiator Corp., 55 West 42nd St., New York, N. Y., and (for mail), 113 Chester Ave., Bloomfield, N. J.
HODEAUX, W. L. (*M* 1931), Owner (for mail), W. L. Hodeaux Plbg. & Htg. Co., 215-17 N. Flagler Dr., and 310 Tenth St., West Palm Beach, Fla.

HODGDON, Harry A. (*M* 1919), 153 Norfolk St., Wollaston, Mass.
HOERSTING, Frank J. (*M* 1921), Pres. (for mail), The Hoersting & Holtmann Co., 1133 W. Third St., and 2045 Philadelphia Dr., Dayton, Ohio.
HOFFMAN, Charles S. (*M* 1924), Vice-Pres. (for mail), Baker Smith & Co., Inc., 576 Greenwich St., and 25 Central Park West, New York, N. Y.
HOFFMAN, James D.* (*M* 1903), (*Presidential Member*), (Pres., 1910; 1st Vice-Pres., 1908; Bd. of Governors, 1911-1912), Prof. of Practical Mechanics, Head of Dept., Director of Practical Mech. Lab. (for mail), Purdue University, and 323 University St., W. Lafayette, Ind.
HOFT, Paul J. (*M* 1925; *A* 1924), (for mail), 245 S. Eighth St., and 1119 Wyoming Ave., Philadelphia, Pa.
HOGAN, Edward L.* (*M* 1911), Consulting Engr. (for mail), American Blower Co., 6000 Russell St., and 700 Seward Ave., Detroit, Mich.
HOLBROOK, Frank M.* (*M* 1923), Engr. (for mail), Congoleum-Nairn, Inc., Kearny, and 52 Walnut Crescent, Montclair, N. J.
HOLLADAY, W. L. (*A* 1933), Engrg. Mgr. (for mail), The George Belsey Co., Ltd., 406 Architects Bldg., Los Angeles, and 110 Loma Alta Dr., Altadena, Calif.
HOLLISTER, Edmund W. (*J* 1931), Supt. Plbg. Htg. Vtg. Div., Hudson Falls Hardware Co., 186 Main St., and (for mail), 88 Oak St., Hudson Falls, N. Y.
HOLLISTER, Norman A. (*M* 1933), 7101 Colonial Rd., Brooklyn, N. Y.
HOLT, James (*M* 1933), Asst. Prof. (for mail), Massachusetts Institute of Technology, Cambridge, and 1062 Massachusetts Ave., Lexington, Mass.
HOLTON, John H. (*M* 1927), Director of Operations (for mail), Carrier Research Corp., 750 Frelinghuysen Ave., Newark, and 30 Hickory Dr., Maplewood, N. J.
HOOD, O. P. (*Honorary Member* 1929), (for mail), Chief, Technologic Br., U. S. Bureau of Mines, 17th and F St. N.W., and 1831 Irving St. N.W., Washington, D. C.
HOOPER, Vernon F. (*M* 1929), 61 Eastern Ave., Ossining, N. Y.
HOPPER, Garnet H. (*M* 1923), Engr., Taylor Forbes, Ltd., 1088 King St. W., and (for mail), 19 Brummell Ave., Toronto, Ont., Canada.
HOPSON, William T. (*M* 1915), The Hopson & Chapin Mfg. Co., New London, Conn.
HORNUNG, John C. (*M* 1914), Engr. (for mail), Central Heat Appliances, 343 S. Dearborn St., Chicago, and 854 Bluff St., Glencoe, Ill.
HORTON, George H. (*M* 1930), 4938 Rosewood Ave., Los Angeles, Calif.
HORTON, Homer F. (*M* 1925), Sales Repr. (for mail), Natl. Regulator Co., 2301 Knox Ave., Chicago, and 343 Green Bay Rd., Glencoe, Ill.
HOSHALL, Robert H. (*M* 1930), Associate (for mail), Thos. H. Allen, Consulting Engr., 65 McCall St., and 789 N. Evergreen St., Memphis, Tenn.
HOSKING, Homer L. (*M* 1930), Br. Sales Mgr. (for mail), Pacific Steel Boiler Corp., 370 Lexington Ave., New York, and 5 Church Lane, Scarsdale, N. Y.
HOSTERMAN, Charles O. (*M* 1924), 25 Bateswell Rd., Dorchester, Mass.
HOTCHKISS, Charles H. B. (*M* 1927), Heating and Ventilating, 148 Lafayette St., New York, N. Y.
HOUGHTEN, Ferry C.* (*M* 1921) Director (for mail), Research Lab., A.S.H.V.E., U. S. Bureau of Mines, 4800 Forbes St., and 1136 Murray Hill Ave., Pittsburgh, Pa.
HOULISTON, G. Baillie (*A* 1928), Secy. (for mail), W. C. Green Co., 704 Race St., Cincinnati, Ohio, and 112 Forest Ave., Ft. Thomas, Ky.
HOWATT, John* (*M* 1915), (Council, 1927-1933), Chief Engr. (for mail), Bd. of Education, 228 N. LaSalle St., and 4940 East End Ave., Chicago, Ill.
HOWELL, Frank B. (*M* 1920), Tech. Advisor (for mail), American Radiator Co., 40 West 40th St., and 15 Central Park West, New York, N. Y.
HOWELL, Lloyd (*M* 1915), Consulting Engr., Htg., Vtg., Cooling, 2239 W. Lake St., and (for mail), 7601 Yates Ave., Chicago, Ill.
HOYT, Charles W. (*A* 1931), Pres-Treas. (for mail), Wolverine Htg. and Vtg. Equip. Co., 80 Boylston St., Boston, and 45 Thaxter Rd., Newtonville, Mass.
HOYT, Leroy W. (*M* 1930), N. Stamford Ave., Stamford, Conn.
HOYT, William B.* (*M* 1919), The Hoyt-Grant Co., 52 Whitney Ave., New Haven, Conn.
HUBBARD, George Wallace* (*M* 1911), Chief Mech. Engr. (for mail), Graham, Anderson, Probst & White, 1417 Railway Exchange, Chicago, and 710 Bonnie Brae, River Forest, Ill.
HUCH, A. J. (*M* 1919), Secy-Treas. (for mail), Central Supply Co., 312 S. Third St., and 4037 Harriet Ave., Minneapolis, Minn.
HUCKER, Joseph H. (*M* 1921), Partner, 1700 Walnut St., Philadelphia, and (for mail), 715 Stranbridge St., Norristown, Pa.
HUFFAKER, Herbert B. (*M* 1933), Htg. Engr. (for mail), Dail Steel Products Co., Lansing, Mich., and 1106 Fourth Ave., Council Bluffs, Iowa.
HULL, Harry B. (*M* 1931), Mgr. Research Eng., Frigidaire Corp., and (for mail), P. O. Box 671 Dayton, Ohio.
HUMPHREY, Dwight E.* (*M* 1921), Htg. and Vtg. Engr. (for mail), Goodyear Tire & Rubber Co., 1144 E. Market St., Akron, and 2499 Sixth St., Cuyahoga Falls, Ohio.
HUMPHREYS, Clark M. (*M* 1931), Asst. Prof. (for mail), Carnegie Institute of Technology, Schenley Park, and 932 Flemington Ave., Pittsburgh, Pa.
HUMPHREYS, James (*M* 1930), Sharp Hill Rd., Wilton, Conn.
HUNGER, Robert F. (*M* 1927), Associate Dist. Mgr. (for mail), Buffalo Forge Co., 703 Cunard Bldg., and 4618 Chester Ave., Philadelphia, Pa.
HUNGERFORD, Leo (*M* 1930), Pres. (for mail), Pacific Elec. & Mech. Co., Inc., 524 Loew's State Bldg., and 105 N. Berendo St., Los Angeles, Calif.
HUNTER, Verne W. (*M* 1932), Supt. (for mail), 417 Grant Bldg., and 3125 Middletown Rd., Sheridan Sta., Pittsburgh, Pa.
HURLEY, Joseph C. (*M* 1915), Pres. (for mail), Petroleum Fuel Engrg. Co., Inc., 4028-32 Filbert St., Philadelphia, and 134 Lansdowne Court, Lansdowne, Pa.
HUSBAND, Edward W. (*M* 1922), 9 Savings St., Providence, R. I.
HUST, Carl E. (*M* 1932), Htg. Engr. (for mail), The Union Gas & Elec. Co., Room 1008, Fourth and Main Sts., and Hillcrest Apts., 15 Mason St., Cincinnati, Ohio.
HUSTOEL, Arnold M. (*A* 1930), 2623 N. Ballou St., Chicago, Ill.
HUTZEL, August F. (*M* 1916), Partner (for mail), 119 E. Washington, and 2115 Wallingford Rd., Ann Arbor, Mich.
HUTZEL, Hugo F. (*M* 1918), Mgr., Erskine Radiator Div. (for mail), Chase Brass & Copper Co., Inc., and 72 Hewlett St., Waterbury, Conn.
HVOSLEF, Fredrik W. (*M* 1931; *A* 1921), Htg. Research Engr. (for mail), Kohler Co., and 523 Audubon Rd., Kohler, Wis.
HYMAN, Wallace M. (*M* 1920), Vice-Pres. (for mail), Reis & O'Donovan, Inc., 255 West 28th St., and 23 West 73rd St., New York, N. Y.
HYNES, Lee P.* (*M* 1919), (for mail), 240 Cherry St., Philadelphia, Pa., and Haddonfield, N. J.

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INGALLS, Frederick D. B. (*M* 1906), Consulting Htg. Engr., 1 Hopkins St., Reading, Mass.
INGELS, Margaret* (*M* 1923; *J* 1918), Mech. Engr. (for mail), Carrier Engrg. Corp., 850 Frelinghuysen Ave., Newark, and Hotel East Orange, East Orange, N. J.
INGOLD, John W. (*S* 1933), 3036 Perrysville Ave. N.S., Pittsburgh, Pa.
INNIS, Helen R.* (*M* 1921; *J* 1918), Largent, W. Va.
ISSERTELL, Henry G.* (*M* 1913; *A* 1912), Consulting Engr., 31 Park Terrace W., New York, N. Y.

J

JACKSON, Alton B. (*M* 1932), Carrier Products Corp., 20 Providence St., Boston, and (for mail), 15 Herrick St., Winchester, Mass.
JACKSON, Charles H. (*M* 1923), Vice-Pres. (for mail), Blower Application Co., 918 N. Fourth St., and 2706 N. Farwell Ave., Milwaukee, Wis.
JACKSON, Marshall S. (*M* 1919), Repr. (for mail), Powers Regulator Co., 250 Delaware Ave. and 108 Larchmont Rd., Buffalo, N. Y.
JACOBSON, Ruben A. (*S* 1933), 1626 Sixth St. S.E., Minneapolis, Minn.
JACOBUS, Dr. David S. (*Life Member*; *M* 1916), The Babcock & Wilcox Co., 85 Liberty St., New York, N. Y.
JALONACK, Irwin G. (*A* 1933; *S* 1930), Engrg. Mgr. (for mail), 82 Railroad Ave., and 15 South St., Patchogue, N. Y.
JAMES, Hamilton R. (*M* 1931), Service Equip. Engr., United Engineers & Constructors, Inc., 1401 Arch St., Philadelphia, and (for mail), 55 W. Drexel Ave., Lansdowne, Pa.
JAMES, John W. (*J* 1933), Research Fellow in Heat. & Vent. (for mail), Mech. Engrg. Bldg., University of Wisconsin, and 1905 Commonwealth Ave., Madison, Wis.
JANCO, Nathan (*S* 1932), Asst. in Mech. Engrg. New York University, and (for mail) 2476 Grand Ave., New York, N. Y.
JANET, Harry L. (*M* 1920), Engr. (for mail), Carrier Engrg. Corp., Chrysler Bldg., New York, and 688 Decatur St., Brooklyn, N. Y.
JARDINE, Douglas C. (*M* 1929; *A* 1926), Pres. (for mail), Jardine & Knight Plbg. & Htg. Co., 312 N. Custer Ave., and 1326 N. Wahsatch Ave., Colorado Springs, Colo.
JARRETT, Paul R. (*A* 1931), 117 Fifth Ave. N., Nashville, Tenn.
JELLETT, Stewart A.* (*Honorary Member*, 1929), (*Charter Member; Presidential Member*), (Pres., 1895; Bd. of Mgrs., 1896-1899; Secy., 1898), Pres. (for mail), Stewart A. Jellett Co., 1200 Locust St., Philadelphia, and 6701 Lincoln Dr., Mt. Airy, Philadelphia, Pa.
JENNEY, Hugh B. (*A* 1933), 1322 Dufferin St., Toronto, Ont., Canada.
JENNINGS, Irving C. (*M* 1924), Nash Engrg. Co., South Norwalk, Conn.
JENNINGS, Warren G. (*A* 1930), Minneapolis-Honeywell Regulator Co., 2065 Daily News Bldg., Chicago, Ill.
JENNINS, Henry H. (*Life Member; M* 1901), E. Oldroyd & Co., Ltd., Black Bull St., Leeds, England.
JENS, Jens (*J* 1930), Mech. Engr. (for mail), Carrier Lufttechnische Gesellschaft, Archiv. Strasse 14, Stuttgart, Germany.
JENSON, Jean S. (*M* 1912), Consulting Engr. (for mail), Neiler-Rich & Co., 431 S. Dearborn St., and 1634 West 106th St., Chicago, Ill.
JOHN, Victor P. (*M* 1931), 136 Berryman Dr., Snyder, N. Y.
JOHNS, Harold B.* (*M* 1928; *J*1927), (for mail), Peoples Gas Co., 122 S. Michigan Ave., Chicago, and 543 N. Elmwood Ave., Oak Park, Ill.
JOHNSON, Carl E. (*J* 1930), Holland Furnace Co., 2001 N. Broad St., Philadelphia, Pa.
JOHNSON, Carl W. (*M* 1912), Pres. (for mail), C. W. Johnson, Inc., 211 N. Desplaines St., and 1809 Morse Ave., Chicago, Ill.
JOHNSON, Clarence W. (*M* 1933; *A* 1933; *J* 1931), Br. Mgr. (for mail), Canadian Sirocco Co., Ltd., 630 Dorchester St. W., and 5549 Queen Mary Rd., Montreal, P. Q., Canada.
JOHNSON, Edward B. (*M* 1919), Sales Engr., Staten Island Supply Co., and (for mail), 154 Wardwell Ave., West New Brighton, S. I., N. Y.
JOHNSON, Edgar E. (*M* 1926), Sales Engr. (for mail), Buffalo Forge Co., 490 Broadway, and 103 University Ave., Buffalo, N. Y.
JOHNSON, Helge S. (*A* 1933; *J* 1927), Buffalo Forge Co., 414 Standard Bldg., Albany, N. Y.
JOHNSON, James M. (*A* 1928), Apt. 207, 3446 Connecticut Ave. N.W., Washington, D. C.
JOHNSON, Leslie O. (*J* 1930), 3307 Berteau Ave., Chicago, Ill.
JOHNSON, Louis H. (*M* 1931), 918 LaSalle Ave., Minneapolis, Minn.
JOHNSON, Paul H. (*A* 1914), E. H. Sheldon & Co., Muskegon, Mich.
JOHNSTON, J. Ambler (*M* 1912), Partner (for mail), Carneal, Johnston & Wright, 806 Electric Bldg., and Richmond, Va.
JOHNSTON, John F., Jr. (*A* 1932), (for mail), Johnston Bros., Inc., Ferrysburg, and Grand Haven, Mich.
JOHNSTON, Robert Elliott (*M* 1929; *A* 1926), 3342–33rd Ave. W., Vancouver, B. C., Canada.
JOHNSTON, William H. (*M* 1924), 306 East 20th St., New York, N. Y.
JONES, Alfred (*M* 1928), Chief Consulting Engr. (for mail), Armstrong Cork Co., P. O. Box 540, and 402 President Ave., Lancaster, Pa.
JONES, Alfred L. (*M* 1926), Plbg. & Htg. Contractor (for mail), 21 Church St., Greenwich, and Box 121, Riverside, Conn.
JONES, Bernard G. (*M* 1928), Mgr. (for mail), Acme Fan & Blower Co., Ltd., 868 Arlington St., and 542 Raglan Rd., Winnipeg, Man., Canada.
JONES, Charles R. (*A* 1928), Jones Supply Co., Siloam Springs, Ark.
JONES, Edwin (*M* 1933; *J* 1924), Box 582, Tulsa, Okla.
JONES, Edwin A. (*M* 1919), C. E. (for mail), L. J. Mueller Furnace Co., 2001 W. Oklahoma, and Shorecrest Hotel, Milwaukee, Wis.
JONES, Edwin F. (*M* 1923), Consulting Engr. (for mail), 420 New York Bldg., and 220 Montrose Pl., St. Paul, Minn.
JONES, Harold L. (*M* 1920), Asst. Supt. (for mail), W. W. Farrier Co., 44 Montgomery St., Jersey City, and 11 Cambridge Rd., Glen Ridge, N. J.
JONES, Noel W. (*S* 1933), 1315 East 28th St., Minneapolis, Minn.
JONES, Raymond E. (*M* 1919), Asst Supervisor Fuel Oil Sales, Gulf Refining Co., 1515 Locust St., Philadelphia, Pa., and (for mail), 39 West End Ave., Haddonfield, N. J.
JONES, William T. (*M* 1915), (Pres., 1933; 1st Vice-Pres., 1932; 2nd Vice-Pres., 1931; Council, 1925-1933), Treas. (for mail), Barnes & Jones, 128 Brookside Ave., Jamaica Plain, and 1886 Beacon St., Waban, Mass.
JORDAN, Richard C. (*S* 1933), 2518 Grand Ave. S., Minneapolis, Minn.
JOYCE, Harry B. (*M* 1922), Consulting Engr. (for mail), 810 Commerce Bldg., and 501 Liberty St., Erie, Pa.
JUNG, John S. (*M* 1930; *A* 1923), Htg. Contractor (for mail), 2409 W. Greenfield Ave., and 1516 S. Layton Blvd., Milwaukee, Wis.
JUTTNER, Otto J. (*M* 1915), Pres. (for mail), Juttner Htg. Co., 814 N. Milwaukee St., and 910 E. Wells St., Milwaukee, Wis.

K

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KAGEY, Isaac B., Jr. (*J* 1929), Metropolitan Life Insurance Co., 1 Madison Ave., New York, N. Y.
KAHAN, Charles (*S* 1933), 6 Marie Ave., Cambridge, Mass.

KAMMAN, Arnold R. (*A* 1925; *J* 1921), (for mail), Chippewa Plbg. Co., Inc., 279 Chicago St., Buffalo, and R. F. D. 1, Hamburg, N. Y.
KAPPEL, George W. A. (*M* 1921), Pres-Treas. (for mail), Camden Htg. Co., Wilson Blvd. and Waldorf Ave., Camden, and 347 W. Kings Highway, Haddonfield, N. J.
KARGES, Louis (*S* 1933), 605 E. Pearl St., Butler, Pa.
KARLSON, Alfred F. (*M* 1918), Chief Engr. (for mail), Parks-Cramer Co., 970 Main St., Fitchburg, and 186 Prospect St., North Leominster, Mass.
KARTORIE, V. T. (*S* 1933), 2982 East 102nd St., Cleveland, Ohio.
KASTNER, George C. (*S* 1933), 654 East 226th St., New York, N. Y.
KATSUMOTO, Eijiro (*M* 1926), Katsumoto & Co., Ginza St., Dairen, Manchuria, China.
KAUFMAN, William M. (*S* 1933), 2875 Sedgwick Ave., New York, N. Y.
KEEFE, Edmund T. (*M* 1931), 75 Pitts St., Boston, Mass.
KEELER, Anson H. (*M* 1932), Sales Engr. (for mail), Vento Div., American Radiator Co., 2212 Walnut St., Philadelphia, and 906 Concord Ave., Drexel Hill, Pa.
KEELING, Harry B. (*M* 1930; *A* 1930), 305 Union Insurance Bldg., Los Angeles, Calif.
KEENEY, Frank P. (*A* 1915), Pres., Engrg. Publications, Inc., 1900 Prairie Ave., Chicago, Ill.
KEHM, Horace Stevens (*M* 1928), 51 E. Grand Ave., Chicago, Ill.
KEIST, Walter E. (*A* 1931), Engr., 393 Center Ave., West View, Pittsburgh, Pa.
KELBLE, Frank R. (*M* 1928), Vice-Pres. and Mgr. (for mail), Huffman-Wolfe Co. of Philadelphia, 11 W. Rittenhouse St., Philadelphia, and 115 Rosyln Ave., Glenside, Pa.
KELLEY, James J. (*A* 1924), Vice-Pres. and Gen. Mgr. (for mail), Arthur H. Ballard, Inc., 535 Commonwealth Ave., Boston, and 142 Governors Ave., Medford, Mass.
KELLNER, Day C. (*S* 1933), Carnegie Institute of Technology, Pittsburgh, Pa.
KELLOGG, Alfred (*M* 1916), (Council, 1920-1921; 1923-1924), Consulting Engr. (for mail), 585 Boylston St., Boston, and 6 Hawthorne St., Belmont, Mass.
KELLY, Charles J. (*M* 1931), New York Repr., Jas. P. Marsh Corp., 551 Fifth Ave., New York, N. Y., and (for mail), 162 Fairview Ave., Jersey City, N. J.
KELLY, Hugh (*M* 1927), Mgr. (for mail), H. Kelly & Co., Ltd., 10041-101 A Ave., and 10235-124th St., Edmonton, Alta., Canada.
KELLY, John G. (*A* 1919), 374 Park Ave., Yonkers, N. Y.
KELLY, Joseph A. (*A* 1931), 3042 N. Franklin St., Philadelphia, Pa.
KENDALL, Edwin H. (*A* 1932; *J* 1930), Engr. (for mail), English-Lauer, Inc., 1224 S. San Pedro, Los Angeles, and 1470 Poppy Peak, Pasadena, Calif.
KENNEDY, Maron (*J* 1930), Sales Engr. (for mail), York Ice Machinery Corp., 5051 Santa Fe Ave., Los Angeles, and 3465 Walnut St., Huntington Park, Calif.
KENNEDY, Owen A. (*S* 1933), 1051 Morewood Ave., Pittsburgh, Pa.
KENT, J. King (*J* 1928), Pres. (for mail), J. King Kent & Co., Inc., 124 N. Main St., St. Louis, and 1030 Commodore Dr., Richmond Heights, Mo.
KENT, Laurence F. (*A* 1927; *J* 1924), Pres. (for mail), Moncrief Furnace Co., P. O. Box 1673, Atlanta, and R. F. D. No. 2, Smyrna, Ga.
KENWARD, Stanley B. (*S* 1933), 45 Fifth Ave., Bay Shore, N. Y.
KEPLINGER, William L. (*M* 1929), Special Repr. (for mail), Carrier Engrg. Corp., 408 Chrysler Bldg., New York, and 103 Sunset Dr., Hempstead, L. I., N. Y.
KEPPNER, Harry W. (*M* 1930), (for mail), H. W. Keppner, 1310 South 56th Ave., and 1245 S. Austin Blvd., Cicero, Ill.
KERN, Raymond T. (*M* 1927), Chief Engr., Jennison Co., Fitchburg, and (for mail), 51 Claflin St., Leominster, Mass.
KERNEY, Thomas F. (*A* 1930; *J* 1925), Engr., H. B. Hackett Consulting Engr., 901 Architects Bldg., and (for mail), 4524 N. Reese St., Philadelphia, Pa.
KERSHAW, Melville G. (*M* 1932; *A* 1926; *J* 1921), Vtg. and Air Cond. Engr. (for mail), E. I. Du Pont de Nemours & Co., Wilmington, Del., and 7313 North 21st St., Philadelphia, Pa.
KEYES, Robert E. (*M* 1913), Chief Engr., The Cooling & Air Conditioning Corp., Hyde Park, Mass.
KICE, M. S., Jr. (*M* 1920), Asst. Chief Engr. (for mail), American Blower Corp., 6000 Russell St., Detroit, and Glengarry Rd., Bloomfield Village near Birmingham, Mich.
KIEFER, Carl J. (*M* 1922), Consulting Engr. (for mail), 918 Schmidt Bldg., and 984 Lennox Pl., Cincinnati, Ohio.
KIEFER, E. J., Jr. (*A* 1932; *J* 1928), Treas. and Gen. Mgr. (for mail), H. C. Archibald Co., 8 S. Sixth St., and 108 N. Sixth St., Stroudsburg, Pa.
KIESLING, Justin A. (*M* 1930), Pres. (for mail), Robischung Kiesling, Inc., 1609 Congress Ave., P. O. Box 1295, and 1806 Holman Ave., Houston, Texas.
KIEWITZ, Arthur A. (*M* 1912), 23-80-28th St., Astoria, L. I., N. Y.
KILNER, John S. (*M* 1929), Sales Engr. (for mail), Kilner-Mills Co., 427 Stormfeltz-Loveley Bldg., and 1091 Seminole Ave., Detroit, Mich.
KIMBALL, Charles W. (*M* 1915), Richard D. Kimball Co., 6 Beacon St., Boston, Mass.
KIMBALL, Dwight D.* (*M* 1908), (*Presidential Member*), (Pres., 1915; 2nd Vice-Pres., 1914; Bd. of Governors, 1912-1916), Consulting Engr. (for mail), 205 East 42nd St., and 21 East 90th St., New York, N. Y.
KING, Roy L. (*S* 1933), 2538 Clinton Ave. S., Minneapolis, Minn.
KINGSLEY, Edwin A. (*M* 1926), 370 Lexington Ave., New York, N. Y.
KINNER, James E. (*M* 1924), The Bryant Heater & Mfg. Co., 17825 St. Clair Ave., Cleveland, Ohio.
KINNEY, William H. (*M* 1931), Bureau of Subways, 20 N. Wacker Dr., Chicago, Ill.
KINTZ, Leslie H. (*M* 1930), Htg. Engr., 210 Highland Ave., Highland Park, Mich.
KIPE, J. Morgan (*M* 1919), 801 Homestead Ave., Beechwood, Del. Co., Pa.
KIRK, Charles D. (*M* 1909), Mgr. (for mail), Chas. D. Kirk Co., Sargent and Colleen, and 774 McMillan Ave., Winnipeg, Man., Canada.
KIRK, Leonard G. (*M* 1923), Kirk & Story, 12 Waverly Pl., New York, N. Y.
KIRKPATRICK, Arthur H. (*J* 1931), Ilg Electric Vtg. Co., 2850 N. Crawford Ave., and (for mail), 4251 Irving Park Blvd., Chicago, Ill.
KITAURA, Shigeyuki (*M* 1918), 191 Shimo-ohsaki near Tokyo, Japan.
KITCH, Stanley B. (*M* 1928; *A* 1928; *J* 1925), 2024 Berwyn Ave., Chicago, Ill.
KITCHELL, Herbert N. (*A* 1926), 4528 Circle Ave., Cincinnati, Ohio.
KITCHEN, Francis A. (*A* 1927; *J* 1923), Pres. (for mail), American Warming & Vtg. Co., 1514 Prospect Ave., Cleveland, and 3552 Glencairn, Shaker Heights, Cleveland, Ohio.
KITCHEN, John H. (*M* 1906), Owner (for mail), John H. Kitchen Co., 1016 Baltimore Ave., and 5015 Westwood Ter., Kansas City, Mo.
KLEIN, Albert (*M* 1920), Managing Dir. (for mail), Carrier Lufttechnische Gesellschaft mbH, Stuttgart, Archivstrasse 14/16 and Stuttgart, Panoramastrasse 23, Germany.
KLEIN, Edward W. (*M*1917), S. E. Dist. Mgr. (for mail), Warren Webster & Co., 152 Nassau St. N.W., and 456 Peachtree Battle Ave., Atlanta, Ga.
KLIE, Walter (*M* 1915), Pres. (for mail), The Smith & Oby Co., 6107 Carnegie Ave., Cleveland, and 18411 S. Woodland Ave., Shaker Heights, Ohio.

MAGINN, Peter F. (*Life Member; M* 1908), Mfgrs. Agent, 1140 S. Negley Ave., Pittsburgh, Pa.

MaGIRL, Willis J. (*A* 1931; *J* 1927), Chief Engr. (for mail), P. H. McGirl Foundry & Furnace Works, 401-13 E. Oakland Ave., and 1528 N. Clinton Blvd., Bloomington, Ill.

MAGNEY, Gottlieb R. (*M* 1931), Pres. (for mail), Magney & Tusler, Inc., Archts. and Engrs., 104 S. Ninth St., and 5329 Washburn Ave. S., Minneapolis, Minn.

MAHONEY, David J. (*M* 1930; *A* 1926), Br. Mgr. (for mail), Johnson Service Co., 503 Franklin St., and 99 Delham Ave., Buffalo, N. Y.

MAIER, George M. (*M* 1921), American Radiator Co., 8007 Jos Campau, Detroit, Mich.

MAIER, Herman F. (*M* 1926), 7124 Morgan St., Chicago, Ill.

MAIMAN, Herbert (*S* 1932), 79-04–78th Ave., Glendale, L. I., N. Y.

MALLIS, William (*M* 1914), 330 Lyon Bldg., Seattle, Wash.

MALONE, Dayle G. (*M* 1929; *A* 1925), 7315 Merrill Ave., Chicago, Ill.

MALVIN, Ray C. (*M* 1929), Pres. (for mail), Malvin & May, Inc., 332 S. Michigan Ave., and 8211 Langley Ave., Chicago, Ill.

MANDEL, Henry J. (*A* 1929), Indiana Gas Utilities Co., Cherry St., Terra Haute, Ind.

MANDEVILLE, Edgar W. (*M* 1914), 1171 East 37th St., Brooklyn, N. Y.

MANN, Arthur R. (*M* 1930), Archt., Mann & Co., Archts. and Engrs., 721 R.W. Bldg., Hutchinson, Kan.

MANN, Lee B. (*J* 1930), Engr. (for mail), Carrier Engrg. Corp., 12 South 12th St., and 4527 Walnut St., Philadelphia, Pa.

MANNING, Walter M. (*M* 1930), P. O. Box 365, Clarks, Nebr.

MARINO, Dominic A. (*S* 1933), New York University, University Heights, and (for mail), 209 East 110th St., New York, N. Y.

MARKS, Alexander A. (*A* 1930), 3441 W. Queen Lane, Philadelphia, Pa.

MARKUSH, Emery U. (*M* 1931), Mech. Engr., Secy. (for mail), Weitbeer Plbg. Corp., 225 East 21st St., New York, and 8601–85th St., Woodhaven, L. I., N. Y.

MARSCHALL, Peter J. (*M* 1930; *A* 1930; *J* 1927), Secy. (for mail), E. Vernon Hill Co., 121 N. Clark St., and 8228 Langley Ave., Chicago, Ill.

MARSHALL, H. Hall (*M* 1923), 37 West 43rd St., New York, N. Y.

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MARTIN, Albert B. (*M* 1917), Kewanee Boiler Co., 1858 S. Western Ave., Chicago, Ill.

MARTIN, George W.* (*M* 1911) Supervising Engr. (for mail), U. S. Realty & Improvement Co., 597 Madison Ave., New York, N. Y., and 340 Prospect St., Ridgewood, N. J.

MARTINEZ, Juan J. (*J* 1929), Paseo de la Reforma 183, Mexico, D. F.

MARTY, Edgar O. (*M* 1916), Pres. and Gen. Mgr., Sherman Coal Corp., and (for mail), 1775 Howard Ave., Pottsville, Pa.

MARUM, Otto (*M* 1931), Plant Engr. (for mail), Agfa Ansco Corp., 29 Charles St., and 12 Grand Blvd., Binghamton, N. Y.

MATCHETT, James C. (*M* 1923), Vice-Pres. and Gen. Mgr. (for mail), Illinois Engrg. Co., 21st and Racine Ave., and 9936 S. Winchester Ave., Chicago, Ill.

MATHER, Harry H. (*A* 1929), Philadelphia-Electric Co., 1000 Chestnut St., Philadelphia, Pa.

MATHEY, Nicholas J. (*M* 1915), Mathey Plbg. & Htg. Co., 31 Third Ave. N.E., Le Mars, Iowa.

MATHIS, Eugene* (*M* 1922), New York Blower Co., 32nd St. and Shields Ave., Armour P. O. Sta., Chicago, Ill.

MATHIS, George A. (*A* 1931), Asst. Supt. (for mail), New York Blower Co., and 108 Highland Court, La Porte, Ind.

MATHIS, Henry (*M* 1921), The New York Blower Co., 32nd and Shields Ave., and (for mail), 10317 Oakley Ave., Chicago, Ill.

MATHIS, Julian W. (*A* 1921), New York Blower Co., 32nd St. and Shields Ave., Chicago, Ill.

MATHIS, Victor John (*S* 1933), 11307 S. Longwood Dr., Chicago, Ill.

MATZEN, Harry B. (*M* 1919), Vice-Pres. (for mail), Carrier Engrg. Corp., 890 Union Trust Bldg., Cleveland, and 3115 Chadbourne Rd., Shaker Heights, Ohio.

MATULLO, Joseph R. (*S* 1933), 485 North 13th St., Newark, N. J.

MAUER, William J.* (*M* 1919), Sales Mgr., Unit Heater Div. (for mail), C. A. Dunham Co., 450 E. Ohio St., Chicago, and 2525 Colfax St., Evanston, Ill.

MAURER, Edward D. (*M* 1921), 1527 Mars Ave., Lakewood, Ohio.

MAUTSCH, Robert (*A* 1928), Engr., Managing Dir. (for mail), Compagnie Belge Des Freins Westinghouse 97 Avenue Louise, and Avenue des Klauwaerts 38 Brussels, Belgium.

MAXWELL, George W. (*S* 1932), Owner, Kenealy & Maxwell Co., and (for mail), Harwich Port, Mass.

MAY, Clarence W. (*M* 1933), Pres. (for mail), May & Griffin, Inc., Consulting Engrs., 320 Perkins Bldg., and 711 N. L St., Tacoma, Wash.

MAY, Edward M. (*M* 1931), 1814 Longview Dr., Springfield, Ohio.

MAY, George Elmer (*M* 1933), Air Cond. Engr. (for mail), New Orleans Public Service, Inc., 317 Baronne St., and 2031 Short St., New Orleans, La.

MAY, Maxwell F. (*M* 1929), Malvin & May, 332 S. Michigan Ave., Chicago, Ill.

MAYNARD, J. Earle (*M* 1931), Chief Htg. Engr., Fox Furnace Co., and (for mail), Telegraph Rd., Elyria, Ohio.

McCARTHY, Charles J. (*M* 1919), Plbg. and Htg. Contractor, 3419 Haverford Ave., Philadelphia, Pa.

McCAULEY, James H. (*M* 1921), James H. McCauley, Inc., 5321 West 65th St., Chicago, Ill.

McCLANAHAN, Luther C. (*M* 1930), 1113 Lancaster St., Pittsburgh, Pa.

McCLELLAN, James E. (*M* 1922), Mgr. (for mail), American Blower Corp., 228 N. LaSalle St., Chicago, and 4741 Washington St., Niles Center, Ill.

McCLELLAND, Harvey S. (*A* 1930), 1928 S. Compton Ave., Los Angeles, Calif.

McCLINTOCK, Alexander, Jr. (*M* 1928; *J* 1920), Member of Firm (for mail), A. McClintock & Sons, 1937 Ridge Ave., and 121 Rochelle Ave., Wissa, Philadelphia, Pa.

McCOLL, Jay R.* (*M* 1916), (*Presidential Member*), (Pres., 1922; 1st Vice-Pres., 1921; 2nd Vice-Pres., 1920; Council, 1920-1923), 2304 Penobscot Bldg., Detroit, Mich.

McCONACHIE, Lorne L. (*A* 1928), Htg. and Plbg., 8817 Mack Ave., and (for mail), 1379 Maryland Ave., Detroit, Mich.

McCONNER, Charles R. (*A* 1925; *J* 1922), Gen. Sales Mgr. (for mail), Clarage Fan Co., and 1904 Waite Ave., Kalamazoo, Mich.

McCORMACK, Denis (*M* 1933), Mgr., Air Cond. Instruments Dept. (for mail), Julien P. Friez & Sons, Inc., 4 N. Central Ave., and 100 W. University Pkwy., Baltimore, Md.

McCOY, Thomas F. (*M* 1924), Mgr. (for mail), The Powers Regulator Co., 125 St. Botolph St., Boston, and Glen Rd., Wellesley Farms, Mass.

McCREA, Lester W. (*M* 1920), Vance-McCrea Sales Co., West 27th and Sisson Sts., Baltimore, Md.

McCREERY, Hugh J. (*M* 1922), (for mail), Marine Bldg., and 1617–49th Ave. W., Vancouver, B. C.

McCUNE, Byron V. (*M* 1928), Sales Engr. (for mail), 101 W. Yakima Ave., P. O. Box 385, and 2310 W. Yakima Ave., Yakima, Wash.

McDONALD, Thomas (*A* 1931), Mgr. (for mail), Minneapolis-Honeywell Regulator Co., Ltd., 117 Peter St., and 56 Kingsway, Toronto, Ont., Canada.

McDONNELL, Everett N. (*M* 1923), Pres. (for mail), McDonnell & Miller, 400 N. Michigan Blvd., and 105 E. Delaware, Chicago, Ill.
McELGIN, John W. (*J* 1931), 180 Rowland Park, Cheltenham, Pa.
McELLROY, George S. (*M* 1925), Consulting Engr., 427 Oliver Bldg., Pittsburgh, and (for mail), R. F. D. No. 2, Glenshaw, Pa.
McFARLAND, William P. (*A* 1923), 1258 Pratt Blvd., Chicago, Ill.
McGINNESS, J. E. (*M* 1903), Pres.(for mail), McGinness, Smith & McGinness Co., 527 First Ave., and 142 Bellfield Ave., Pittsburgh, Pa.
McGLENN, G. Raymond (*M* 1915), 259 Lormore St., Elmira, N. Y.
McGONAGLE, Arthur (*M* 1932), Consulting Engr. (for mail), 1013 Fulton Bldg., Pittsburgh, and 6815 Prospect Ave., Ben Avon, Pa.
McGRAIL, Thomas E. (*M* 1926), 3465 Belmore Ave., Montreal, P. Q., Canada.
McGUIGAN, L. A. (*A* 1919), 724 Hastings St., Pittsburgh, Pa.
McHENRY, Robert W. (*M* 1921), 236 Eglinton Ave. E., Toronto, Ont., Canada.
McILVAINE, John H.* (*M* 1929), 749 Custer Ave., Evanston, and (for mail), McIlvaine Burner Corp., 663 W. Washington Blvd., Chicago, Ill.
McINTIRE, James F. (*M* 1915; *A* 1914), (Council, 1926-1928-1933), Vice-Pres. (for mail), U. S. Radiator Corp., 1056-44 Cadillac Sq., P. O. Box 686, and 3201 Sherbourne Rd., Detroit, Mich.
McINTOSH, Fabian C. (*M* 1921; *J* 1917), (Treas., 1930; Council, 1929-1933), Br. Mgr. (for mail), Johnson Service Co., 1238 Brighton Rd., and 302 Marshall Ave., Pittsburgh, Pa.
McKELVEY, David M. (*M* 1930), 60 East 42nd St., New York, N. Y.
McKIEVER, William H.* (*M* 1897; *J* 1896), Pres. (for mail), William H. McKiever, Inc., 247 West 13th St., New York, and 479 Eighth St., Brooklyn, N. Y.
McLARNEY, Harry W. (*M* 1933), Air Cond. Engr. (for mail), Union Electric Light & Power Co., 315 North 12th St., and 5053 Lindenwood Ave., St. Louis, Mo.
McLAUGHLIN, Joseph D. (*A* 1930; *J* 1928), 166 Aborn St., Providence, R. I.
McLEAN, Dermid (*M* 1917), McColl, Snyder & McLean, 2304 Penobscot Bldg., Detroit, Mich.
McLEISH, William S. (*A* 1932; *J* 1928), Dist. Engr. (for mail), The Ric-Wil Co., Room 1838 101 Park Ave., New York, N. Y., and 1646 Wagar Ave., Lakewood, Ohio.
McLENEGAN, David W. (*M* 1933), Asst. Engr., Air Cond. Dept. (for mail), General Electric Co., and 1086 Rugby Rd., Schenectady, N. Y.
McMURRER, Louis J. (*M* 1928; *A* 1928; *J* 1924), Pres., The McMurrer Co., 303 Congress St., Boston, and (for mail), 190 Harvard Circle, Newtonville, Mass.
McNAIR, Edward E. (*M* 1915), (2nd Vice-Pres., 1923; Council, 1921-1923), Vice-Pres. (for mail), U. S. Radiator Corp., Pres., Pacific Steel Boiler Corp., 1056 First Natl. Bk. Bldg., Detroit, and R. F. D. No. 1, Birmingham, Mich.
McNAMARA, William (*A* 1930), Sales Engr. (for mail), The Trane Co., 722 Plymouth Bldg., Minneapolis, and 1355 Como Ave. W., St. Paul, Minn.
McPHERSON, William A. (*M* 1929), Chief, Htg. and Vtg. Div., Dept. or School Bldgs., 11 Beacon St., Boston, and (for mail), 86 Dwinnell St., West Roxbury, Mass.
McTERNAN, Felix J. (*A* 1931), 1523 Main St., Buffalo, N. Y.
MEAD, Edward A. (*M* 1926), Asst. Sales Mgr. (for mail), Nash Engrg. Co., South Norwalk, and 5 Thames St., Norwalk, Conn.
MEFFERT, George H. (*J* 1930), Engr. (for mail), Carrier Engrg. Corp., 2022 Bryan St., and 4154½ Prescott Ave., Dallas, Texas.
MEHNE, Carl A. (*M* 1929), Htg. and Vtg. Expert (for mail), Room 821, 101 Park Ave., New York, N. Y., and Livingston St., Valhalla, N. Y.
MEINKE, Howard G. (*M* 1933), Asst. Engr., Civil Engrg. Dept. (for mail), New York Edison Co., 4 Irving Pl., Room 1517-S, New York, N. Y., and 41, Harte St., Baldwin Nassau Co., N. Y.
MEISEL, Carl L. (*J* 1931), 350 Central Park W., New York, N. Y.
MELICHAR, Wolfgang J. (*J* 1932), Johannes Haag VII, Neustiftgasse 98, and (for mail), XII, Thunhofgasse 6, Vienna, Austria.
MELLON, James T. J. (*M* 1911), (Council, 1915), (for mail), Mellon Co., 4415-21 Ludlow St., and 431 North 63rd St., Philadelphia, Pa.
MENSING, Frederick D. (*M* 1920), (Natl. Treas., 1931-1932; Council, 1931-1932), Consulting Engr. (for mail), Mensing & Co., 12 South 12th St., and 2845 Frankford Ave., Philadelphia, Pa.
MERKEL, Fred P. (*M* 1924), 204–11th Ave., Belmar, N. J.
MERRILL, Carle J. (*M* 1919), C. J. Merrill, Inc., 54 St. John St., Portland, Me.
MERRITT, C. J. (*M* 1925), Dir., Merritt, Ltd., 8 Quai de France, Shanghai, China.
MERTZ, Walter A. (*M* 1919), Kehm Bros., 51 E. Grand Ave., Chicago, Ill.
MERWIN, Gile E. (*M* 1924; *J* 1923), Secy., Rockford Plbg. Supply Co., and (for mail), 316 Ellsworth St., Rockford, Ill.
MEYER, Charles L. (*M* 1930), 198-25 Foothill Ter., Hollis, L. I., N. Y.
MEYER, Frank L. (*M* 1923; *J* 1928), Vice-Pres., The Meyer Furnace Co., and (for mail), 9 Cole Court, Peoria, Ill.
MEYER, Henry C., Jr.* (*M* 1898), (Council, 1915-1916), 101 Park Ave., New York, N. Y.
MEYER, John W. (*A* 1929), Asst. to Mgr., Industrial Sales Dept. (for mail), Philadelphia Electric Co., 1000 Chestnut St., and 5000 Pine St., Philadelphia, Pa.
MICHIE, D. Fraser (*A* 1930), Boiler and Rad. Div., Crane, Ltd., 93 Lombard St., and (for mail), 5B, 553 Wardlaw Ave., Winnipeg, Man., Canada.
MILLAR, Rowland J. (*M* 1925), Mgr. (for mail), Pease Foundry Co., Ltd., 118 King St. S., and 53 Oakmount Rd., Toronto, Ont., Canada.
MILLARD, Junius W. (*M* 1929), Dist. Mgr. (for mail), Carrier Corp., 410 Asylum St., Hartford, and Manchester, Conn.
MILLER, Alan A. (*A* 1926), Mgr. (for mail), National Radiator Corp., 508 Liberty Trust Bldg., Philadelphia, and 731 Cornell Ave., Drexel Hill, Pa.
MILLER, Bruce R. (*A* 1930), 1533 N. W. 25th St., Oklahoma City, Okla.
MILLER, Charles A. (*A* 1917), Salesman (for mail), The H. B. Smith Co., 10 East 41st St., and 2870 Marion Ave., New York, N. Y.
MILLER, Charles W. (*M* 1919; *J* 1908), Pres. (for mail), The Rado Co., 338 S. Second St., Milwaukee, and R. 1, Box 42, Menomonee Falls, Wis.
MILLER, Edgar S. (*M* 1931), 3012 Campbell St., Kansas City, Mo.
MILLER, Floyd A. (*M* 1911), 477 Federal Bldg., Chicago, Ill.
MILLER, Harry M. (*M* 1920), 3938 N. Stowell Ave., Milwaukee, Wis.
MILLER, Henry F. (*A* 1928), Dist. Mgr. (for mail), Keasbey & Mattison Co., Ambler and Ridley Park, Pa.
MILLER, James E. (*M* 1914; *J* 1912), Vice-Pres. (for mail), C. W. Johnson, Inc., 211 N. Desplaines St., Chicago, and 2210 Colfax St., Evanston, Ill.
MILLER, John F. G. (*M* 1916), Vice-Pres. (for mail), B. F. Sturtevant Co., Hyde Park, Boston, and Longwood Towers, Brookline, Mass.
MILLER, Leo B. (*M* 1926), Refrigeration Div. (for mail), Minneapolis-Honeywell Regulator Co., 2753 Fourth Ave. S., and 2010 James Ave. S., Minneapolis, Minn.
MILLER, Prof. Lorin G. (*M* 1933), Prof. Mech. Engrg. (for mail), Dept. of Mech. Engrg., Michigan State College, Engrg. Bldg., and 920 Sunset Lane, East Lansing, Mich.
MILLER, Merl W. (*M* 1932; *A* 1932; *J* 1926), Mgr. of Lab. (for mail), Trane Co., and 308 North 22nd St., LaCrosse, Wis.

MILLER, Robert A.* (*M* 1931), Tech. Sales Engr. (for mail), Pittsburgh Plate Glass Co., 2200 Grant Bldg., Pittsburgh, and 1211 Carlisle St., Tarentum, Pa.
MILLER, Robert T. (*A* 1927), Chief Engr. (for mail), Masonite Corp., 111 W. Washington St., and 1608 Sherwin Ave., Chicago, Ill.
MILLER, Tolbert G. (*A* 1929; *J* 1921), Supt., Htg. and Vtg., and (for mail), 11 N. Second St., Wormleysburg, Pa.
MILLIKEN, James H.* (*M* 1923), Dist. Repr. (for mail), American Air Filter Co., Inc., 20 N. Wacker Dr., Chicago, and 1021 Ridge Ct., Evanston, Ill.
MILLIKEN, Vincent D. (*A* 1930), Sales Mgr. (for mail), Skidmore Corp., St. Joseph and Colfax Ave., Benton Harbor, Mich.
MILLIS, Linn W. (*M* 1918), 3534 Wabash Ave., Kansas City, Mo.
MILWARD, Robert K. (*A* 1920), Mgr. (for mail), U. S. Radiator Corp., 127 Campbell Ave., and 2441 Calvert Ave., Detroit, Mich.
MITCHELL, Charles H. (*M* 1924), Htg. Engr., 179 Thatcher St., Mattapan P. O., Milton, Mass.
MITTENDORFF, E. M. (*M* 1932), Sales Engr., Sarco Co., Inc., 222 N. Bank Dr., Chicago, and (for mail), 2220 Sherman Ave., Evanston, Ill.
MODIANO, René (*M* 1925), 55 Boulevard Beausejour, Paris 16, eme, France.
MOFFITT, Roy M. (*A* 1930), Br. Mgr., L. J. Mueller Furnace Co., 211 W. Wacker Dr., Chicago, Ill.
MOLER, William H. (*M* 1927; *J* 1923), Mech. Sales Engr., R. F. D. No. 1, Box 37-B, Irving, Texas.
MONDAY, Charles E. (*M* 1920), Chas. E. Monday & Co., 1323 Fairmount Ave., Philadelphia, Pa.
MONROE, Meade (*S* 1933), 1228 Southern Blvd., New York, N. Y.
MONROE, Raymond R. (*A* 1929), Nash Engrg. Co., South Norwalk, and (for mail), Kellogg Park, Norwalk, Conn.
MONTGOMERY, Ora C. (*M* 1933), Asst. Supt. of Power (for mail), N. Y. C. R. R., Grand Central Terminal, Room 1842, and 255 West 84th St., New York, N. Y.
MOON, L. Walter (*M* 1915), (Council, 1933), Pres. (for mail), Bradley Htg. Co., 3834 Olive St., and 5006 N. Kingshighway, St. Louis, Mo.
MOORE, Herbert S. (*A* 1923), Mfrs. Agent, 107 Clendenan Ave., Toronto 9, Ont., Canada.
MOORE, Robert E. (*J* 1933), Salesman, Sarco Co., Inc., 183 Madison Ave., New York, and (for mail), 1730 East 46th St., Brooklyn, N. Y.
MOORE, Robert E. (*A* 1928), 744 Dobson St., Evanston, Ill.
MOREAU, Donato (*A* 1932), J. Knox Corbett Lumber Co., Tucson, Ariz.
MORGAN, C. Stanley (*A* 1919), 445 W. Larned St., Detroit, Mich.
MORGAN, Glenn C. (*M* 1911), Partner (for mail), Morgan-Gerrish Co., 307 Essex Bldg., and 4308 Fremont Ave. S., Minneapolis, Minn.
MORGAN, Robert C. (*M* 1915), 314 W. Seymour St., Philadelphia, Pa.
MOREHOUSE, H. Preston (*M* 1933), General Air Cond. Repr. (for mail), Public Service Elec. & Gas Co., 80 Park Pl., Newark, and 85 Halsted St., East Orange, N. J.
MORRIS, Edward J. (*S* 1931), 3414 Gwynn's Falls Pkwy., Baltimore, Md.
MORRIS, Fred H. (*A* 1929), 1233 East 125th St., Cleveland, Ohio.
MORRISON, Chester B. (*M* 1931), Mgr. (for mail), York Shipley, Inc., 21 Jinkee Rd., and 882 Rue Retard, Shanghai, China.
MORSE, Clark T. (*M* 1913), Pres. (for mail), American Blower Corp., 6000 Russell, and 16222 Shaftsbury Rd., Detroit, Mich.
MORTON, Charles H. (*A* 1931), 1106 Sherman St. S.E., Grand Rapids, Mich.
MORTON, Harold S. (*M* 1931), Dist. Mgr., Modern Coal Burner Co., and (for mail), 58 N. Mississippi River Blvd., St. Paul, Minn.
MOSHER, Clarence H. (*A* 1919), C. H. Mosher Co., 423 Ashland Ave., Buffalo, N. Y.
MOSS, Edward (*M* 1920), 1130 Atlantic Ave., Brooklyn, N. Y.
MOTZ, O. Wayne (*M* 1932), Mech. Engr., Samuel Hannaford & Sons, Archts., 1024 Dixie Terminal Bldg., Cincinnati, and (for mail), 2587 Irving Pl., Norwood, Ohio.
MOULDER, Albert W.* (*M* 1917), Mgr. Htg. Power and Industrial Piping Div. (for mail), Grinnell Co., Inc., 260 W. Exchange St., Providence, and Bluff Rd., Barrington, R. I.
MOULTON, David (*M* 1926), 99 Chauncy St., Boston, Mass.
MOWER, William P. (*M* 1924), Warren Webster & Co., 76 Summer St., Boston, Mass.
MUELLER, Harold C. (*A* 1930), Sales Engr. (for mail), Powers Regulator Co., 2720 Greenview Ave., Chicago, and 2720 Lawndale Ave., Evanston, Ill.
MUNDER, John F., Jr. (*M* 1927; *J* 1924), Sales Supervisor (for mail), Carrier Products Corp., Chrysler Bldg., New York, N. Y., and 81 Joyce Rd., Tenafly, N. J.
MUNIER, Leon L. (*M* 1919; *J* 1915), Pres. (for mail), Wolff & Munier, Inc., 222 East 41st St., New York, N. Y., and 63 Columbia Ave., Hartsdale, N. Y.
MUNRO, Edward A. (*Charter Member; Life Member*), Htg. and Vtg. Engr., 56 Jarvis Pl., Lynbrook, N. Y.
MURPHY, Edward T.* (*M* 1915), Pres. (for mail), Carrier Engrg. Corp., 180 N. Michigan Ave., Room 907, and 200 E. Chestnut St., Chicago, Ill.
MURPHY, Howard C.* (*M* 1923), Vice-Pres. (for mail), American Air Filter Co., Inc., 215 Central Ave., and 495 Lightfoot Rd., Louisville, Ky.
MURPHY, Joseph R. (*A* 1925), Riverside Ter., Riverside, Conn.
MURPHY, William W. (*M* 1930), Treas. (for mail), W. W. Murphy Co., 171 Chestnut St., and 25 Mansfield St., Springfield, Mass.
MURRAY, John J. (*A* 1933), Salesman, Vice-Pres., Pierce Perry Co., Boston, and (for mail), 60 Commonwealth Park W., Newton Centre, Mass.
MURRAY, Thomas F. (*M* 1923), State Architect, and (for mail), 14 S. Lake Ave., Albany, N. Y.
MYERS, Eddie V. (*A* 1931), Vice-Pres., A. S. Hunter & Son, Inc., and (for mail), 816 Clark St., Willard, Ohio.
MYERS, Frank L. (*M* 1933), Sales Engr., Owens, Illinois Glass Co., and (for mail), 3406 Detroit Ave., Toledo, Ohio.
MYERS, George W. F. (*M* 1930; *A* 1928; *J* 1923), Mfrs. Repr., Htg., Vtg. and Air Cond. (for mail), Myers Engrg. Equip. Co., Mart Bldg., 401 South 12th St., St. Louis, and 476 Pasadena Ave., Webster Groves, Mo.

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NASON, George L. (*M* 1929; *A* 1929; *J* 1927), 31 N. Franklin St., Holbrook, Mass.
NASS, Arthur F. (*M* 1927), Secy-Treas. (for mail), McGinness, Smith & McGinness Co., 527 First Ave., Pittsburgh, and Elmhurst Rd., R. D. No. 8, Crafton, P.O., Pa.
NATKIN, Benjamin* (*M* 1909; *J* 1907), Gen. Mgr. (for mail), Natkin Engrg. Co., 314 W. Tenth St., and 5211 Rockhill Rd., Kansas City, Mo.
NAYLOR, Charles L. (*M* 1931), Supt., Heat, Light and Power Dept. (for mail), The Atlantic Refining Co., 3144 Passyunk Ave., and 2315 North 18th St., Philadelphia, Pa.
NEAL, Harry W. (*M* 1928), Neal Furnace Co., 2705-7 Northwestern Ave., Indianapolis, Ind.

NEALE, Laurance I. (*A* 1927), Vice-Pres. (for mail), Atlantic Gypsum Products Co., Inc., 60 East 42nd St., and 125 East 57th St., New York, N. Y.
NEARY, Daniel A. (*S* 1933), 444 East 66th St., New York, N. Y.
NEEDLER, J. H. (*M* 1933), Phillips Getschow Co., 32 W. Austin Ave., Chicago, Ill.
NEFF, Charles J. (*S* 1930), Canfield, Ohio.
NEIDECK, Albert A. (*J* 1927), 2134 Wallace Ave., New York, N. Y.
NEILER, Samuel G. (*M* 1898), Consulting Mech. and Elec. Engr. (for mail), Neiler, Rich & Co., 431 S. Dearborn St., Chicago, and 737 N. Oak Park Ave., Oak Park, Ill.
NELSON, Chester L. (*J* 1929), 62 Monona Ave., Rutherford, N. J.
NELSON, D. W.* (*M* 1928), Asst. Prof. of Steam and Gas Engrg. (for mail), Mech. Engrg. Bldg., University of Wisconsin, and 3906 Council Crest, Madison, Wis.
NELSON, George A. (*M* 1928), Sales Engr. (for mail), 112 East 19th St., and 2336 University Ave., New York, N. Y.
NELSON, George O. (*M* 1923), Carstens Bros., Ackley, Iowa.
NELSON, Harold A. (*M* 1926), 236 S. La Pere St., Beverly Hills, Calif.
NELSON, Herman W. (*M* 1909), Pres. (for mail), The Herman Nelson Corp., 1824 Third Ave., and 2500–11th St., Moline, Ill.
NELSON, Richard H. (*A* 1933; *J* 1928), Director, Herman Nelson Corp., 1824 Third Ave., and (for mail), 1303–30th St., Moline, Ill.
NESBITT, Albert J.* (*M* 1921; *J* 1921), Secy-Treas. (for mail), John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Rockfield Farm, Tennis Ave. and Welsh Rd., Ambler, Pa.
NESBITT, John J. (*M* 1923), Pres. (for mail), John J. Nesbitt, Inc., State Rd. and Rhawn St., Philadelphia, and Rockfield Farm, Tennis Ave. and Welsh Rd., Ambler, Pa.
NESDAHL, Eilert (*M* 1915), c/o Colben Nesdahl, Route 1, Shevlin, Minn.
NESMITH, Oliver E. (*A* 1928), 107 Warner Ave., Bloomington, Ill.
NESS, William H. C. (*M* 1931), Gen. Mgr. (for mail), Master Fan Corp., 1323 Channing St., and 215 N. Kingsley Dr., Los Angeles, Calif.
NESSI, André (*M* 1930), Ingr. des Arts et Manufactures, Établissement Nessi Frères & Cie, 43 Rue le la Vanne, Montrouge (Seine), and (for mail), 1 Avenue du President Wilson, Paris VIII, France.
NEU, Henri J. E. (*M* 1933), Pres., Etablissements Neu, 47-49 Rue Fourier, Lille (Nord), France.
NEWMAN, Charles T. (*A* 1931), 2729 Hooper Ave., Los Angeles, Calif.
NEWPORT, Charles F.* (*M* 1906), Sales Engr., Weil-McLain Co., Michigan City, Ind., and (for mail), 10001 Longwood Dr., Chicago, Ill.
NICELY, John E. (*A* 1925), 1208 Marion St., Reading, Pa.
NICHOLLS, Percy* (*M* 1920), Supervising Engr., Fuel Section (for mail), U. S. Bureau of Mines, Pittsburgh, Pa.
NICOL, Norman C. (*M* 1923), P. O. Box 146, Trinity Sta., New York, N. Y.
NIGHTINGALE, George F. (*A* 1931), 621 S. Maple Ave., Oak Park, Ill.
NOBBS, Walter W. (*M* 1919), 50 Fairhazel Gardens, London N.W. 6, England.
NOBIS, Harry M. (*M* 1914), 1827 Stanwood Rd., East Cleveland, Ohio.
NOBLE, Theodore G. (*S* 1933), Engr., Minneapolis Gas Light Co., and (for mail), 523 Oak St., S.E., Minneapolis, Minn.
NOLAN, James J. Jr. (*J* 1929), Carrier Engrg. Corp., 604 Washington Bldg., Washington, D. C.
NOLAND, Lloyd U. (*M* 1915), Noland Co., Inc., Newport News, Va.
NOLL, William F. (*M* 1924), Htg. Contractor, 629 North 27th St., and (for mail), 2850 North 47th St., Milwaukee, Wis.
NORDHEIMER, Clyde L. (*S* 1931), Student, Carnegie Institute of Technology, and (for mail), 622 Mellon St., Pittsburgh, Pa.
NORDINE, Louis F. (*M* 1914), 1170–25th St., Moline, Ill.
NORRIS, William D. (*M* 1930), 1314 Forest Ave., Wilmette, Ill.
NORTHON, Louis (*M* 1929), Consulting Engr., 132 Park Ave., Mt. Vernon, N. Y.
NOTTBERG, Gustav (*A* 1933), Htg. Estimator (for mail), U. S. Engrg. Co., 914 Campbell, and 1835 East 68th St. Ter., Kansas City, Mo.
NOTTBERG, Henry J., Sr. (*M* 1919), Vice-Pres. (for mail), U. S. Engrg. Co., 914 Campbell St., and 3508 Anderson Ave., Kansas City, Mo.
NOVOTNEY, Thomas A. (*M* 1928), Mgr., Research and Sales Engrg. Depts., National Radiator Corp., 221 Central Ave., and (for mail), 403 Wayne St., Johnstown, Pa.
NOWITZKY, Herman S. (*A* 1931), Supt., Construction, Repairs and Maintenance, Wilmer & Vincent Theatrical Circuit, and (for mail), 151 Tenth St., Norfolk, Va.
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OATES, Walter A. (*M* 1931), Htg. and Industrial Engr., Lynn Gas & Electric Co., 90 Exchange St., and (for mail), 285 Lynn Shore Dr., Lynn, Mass.
O'BANNON, Lester S.* (*M* 1928), University of Kentucky, Lexington, Ky.
OBERG, Harry C. (*A* 1933), Mgr. Engrg. Dept., Crane Co., Fifth and Broadway, and (for mail), 1362 W. Minnehaha St., St. Paul, Minn.
OBERT, Casin W.* (*M* 1916), Consulting Engr., Union Carbide & Carbon Research Lab., Thompson Ave and Manley St., Long Island City, and (for mail) 122 N. Columbia Ave., Mt. Vernon, N. Y.
O'BRIEN, J. H. (*M* 1923), 228 N. LaSalle St., Chicago, Ill.
O'CONNELL, Presly M. (*M* 1916), 5749–31st Ave. N.E., Seattle, Wash.
OFFEN, Ben (*M* 1928), Owner (for mail), B. Offen & Co., 608 S. Dearborn St., and 1100 N. Dearborn St., Chicago, Ill.
OFFNER, Alfred J.* (*M* 1922), Consulting Engr. (for mail), 139 East 53rd St., New York, and 160-15–11th Ave., Beechhurst, L. I., N. Y.
O'HARE, George W., Jr. (*S* 1932), Student, New York University, University Heights, and (for mail), 201 West 72nd St., New York, N. Y.
O'HAVER, Hubert M. (*J* 1931), Foss Htg. & Engrg. Co., 21 S. Chester Ave., Pasadena, Calif.
OLCHOFF, Maurice (*M* 1933), Mgr., Olchoff Engrg. Co., 423 Dwight Bldg., and (for mail), 5341 Holmes, Kansas City, Mo.
OLSEN, Carlton F. (*A* 1925; *J* 1920), Kewanee Boiler Corp., and (for mail), 7914 Wabash Ave., Chicago, Ill.
OLSEN, Gustav E. (*M* 1930), 6809 Amstel Blvd., Arverne, L. I., N. Y.
OLSON, Gilbert E. (*M* 1930), 440 Ward Pkwy., Kansas City, Mo.
OLSON, Robert G. (*M* 1923), Sales Engr. (for mail), c/o Hydraulic Coupling Corp., 1349 Harper Ave., and 111 Putnam Ave., Detroit, Mich.
OLSTAD, Martin H. (*A* 1933; *J* 1931), Engr. (for mail), Niagara Blower Co., 6 East 45th St., New York, and 2940–210th Pl., Bayside, L. I., N. Y.
OLVANY, William J. (*M* 1912), Pres. (for mail), William J. Olvany, Inc., 100 Charles St., New York, and 109-40–71st Rd., Forest Hills, L. I., N. Y.

O'NEILL, James W. (*M* 1929; *A* 1927; *J* 1925), Chief Engr., Trane Co. of Canada, Ltd., 439 King St. W., and (for mail), 8 Springmount Ave., Toronto, Canada.
O'NEILL, Peter (*M* 1920), Treas. (for mail), Bartley-O'Neill Co., 240-42 Blvd. of Allies, Pittsburgh, and 2448 Charles St. N.S., Pittsburgh (14), Pa.
OPPERMAN, Everett F. (*S* 1933), 169 Milbank Ave., Greenwich, Conn.
OREAR, Andrew G. (*M* 1930), Sales Engr. and Mfrs. Repr. (for mail), Room 501, San Fernando Bldg., Los Angeles, and 1015 E. Raleigh St., Glendale, Calif.
ORMSBY, H. Kingsley, Jr. (*M* 1930; *A* 1930; *J* 1928), 668 Roberts Ave., Syracuse, N. Y.
ORR, H. B. (*M* 1928), (for mail), "Carrier"-Dravo Doyle Co., 302 Penn Ave., Pittsburgh, and Central Sq. Apt. B-5, Mt. Lebanon, Pa.
OSBORN, Wallace J. (*A* 1927), 599 Old Post Rd., Fairfield, Conn.
OSBORNE, Gurdon H. (*M* 1922), Gen. Mgr., The Vtg. & Blow Pipe Co., Ltd., 714 St. Maurice St., Montreal, and (for mail), 836 Pratt Ave., Outremont, Montreal, Que., Canada.
OSBORNE, Maurice M. (*M* 1925), 367 Beacon St., Boston, Mass.
OSBURN, Richard M. (*S* 1933), 2241 Sedgwick Ave., New York, N. Y.
OSMUNDSEN, Harry B. (*M* 1930), 129 Haddon Pl., Montclair, N. J.
OSTER, George R. (*A* 1930), 921 Hollingsworth Bldg., Los Angeles, Calif.
OTIS, Gerald E.* (*M* 1922), Vice-Pres. (for mail), The Herman Nelson Corp., and 1921–23rd Ave., Moline, Ill.
OTT, Oran W. (*M* 1925), Consulting Mech. Engr. (for mail), Washington Bldg., and 123 S. Virgil Ave., Los Angeles, Calif.
OTT, Rush C. (*M* 1931), Refrigerating Equip. Corp., 927-31 N. Meridian St., Indianapolis, Ind.
OTTO, Robert W. (*M* 1912), 2147 Carroll Ave., St. Paul, Minn.
OURUSOFF, L. (*M* 1931), Engr. of Utilization (for mail), Washington Gas Light Co., 1100–29th St. N.W., Washington, D. C., and 25 W. Irving St., Chevy Chase, Md.
OVERTON, Sidney H. (*M* 1929), 34 Church St., Kensington, London W, 8, England.

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PAETZ, Herbert E. (*M* 1922), Div. Sales Mgr. (for mail), American Blower Corp., 2539 Woodward Ave., and 17568 Roselawn, Detroit, Mich.
PAGE, Harry W. (*M* 1923), Pres., Wisconsin Equipment Co., 204 W. Wisconsin Ave., Milwaukee, and (for mail), 7927 Warren Ave., Wauwatosa, Wis.
PARK, Clifton D. (*M* 1929), 22 Otis St., Needham, Mass.
PARK, J. Frank (*J* 1930), Sales Engr., Air Cond. (for mail), 748 E. Washington Blvd., Los Angeles, and 4142 Linden St., Long Beach, Calif.
PARKER, Philip (*M* 1915), 8 Middle St., Woburn, Mass.
PARKS, William N. (*A* 1927), Br. Mgr. (for mail), U. S. Radiator Corp., 688 Hampden Ave., St. Paul, and 5348 First Ave. S., Minneapolis, Minn.
PARROTT, Lyle George (*M* 1922), Consulting Engr., McColl, Snyder & McLean, 2306 Penobscot Bldg., and (for mail), 4678 Seebaldt Ave., Detroit, Mich.
PARSONS, Roger A. (*J* 1933), Sales Engr., Dail Steel Products Co., and (for mail), 525 W. Grand River Ave., Lansing, Mich.
PARTLAN, James W. (*Life Member; M* 1916), 14290 Goddard Ave., Detroit, Mich.
PATERSON, James S.* (*M* 1922), Mech. Engr. (for mail), Bd. of Education, 155 College St., and 23 Norton Ave., Toronto, Ont., Canada.
PATORNO, Sullivan A. S. (*M* 1923), Chief Draftsman, (for mail), Meyer, Strong & Jones, Inc., 101 Park Ave., and 1269 Findlay Ave., New York, N. Y.
PATRICK, Horace M. (*A* 1933; *J* 1929), Lafayette Rd., Colonial Village, Wayne, Pa.
PAUL, Donald I. (*J* 1932), Sales Engr. (for mail), Gurney Foundry Co., Ltd., 4 Junction Rd., and 222 Fern Ave., Toronto, Ont., Canada.
PAULDING, Lewis G. (*M* 1926), Secy-Treas. (for mail), Frank Paulding & Son, 51 East 42nd St., New York, and 8733–117th St., Richmond Hill, L. I., N. Y.
PEACOCK, James K. (*M* 1921), 440 Fowler Ave., Pelham Manor, N. Y.
PEEBLES, John K., Jr. (*A* 1925; *J* 1924), (for mail), Peebles & Ferguson, 733 Law Bldg., and 1111 W. Princess Anne Rd., Norfolk, Va.
PELLETIER, Albert (*M* 1930), Mech. Engr., Samuel R. Lewis, 407 S. Dearborn St., Chicago, and (for mail), 708 S. Ninth Ave., Maywood, Ill.
PENCE, Millard D. (*A* 1930; *J* 1927), C. A. Dunham Co., 450 E. Ohio St., Chicago, Ill.
PENNEL, Reed (*J* 1933), Buckeye Blower Co., 501 Martin Bldg. N.S., Pittsburgh, Pa.
PENNOCK, William B. (*M* 1927), 170 McNab St. S., Hamilton, Ont., Canada.
PERINA, Arthur E. (*S* 1933), 126 Courtland St., Staten Island, N. Y.
PESTERFIELD, Charles H. (*S* 1932), Ozark, Ark.
PETERS, Herbert H. (*M* 1930), 1842 North 40th St., Milwaukee, Wis.
PETERSEN, Adolph J. (*J* 1929), Sales Engr. (for mail), Philadelphia & Reading Coal & Iron Co., 332 S. Michigan Ave., and 1220 N. State Pkwy., Chicago, Ill.
PETERSON, Sterling D. (*A* 1930), Br. Mgr. (for mail), Johnson Service Co., 473 Colman Bldg., and 5051 Prince St., Seattle, Wash.
PFEIFER, Otto J., Jr. (*J* 1932), Engr., Ralph D. Thomas & Associates, 1200 Second Ave. S., and (for mail), 1515 Monroe St. N.E., Minneapolis, Minn.
PFEIFFER, John F. (*M* 1930; *J* 1925), 346 Louisa St., Williamsport, Pa.
PFUHLER, John L. (*A* 1925; *J* 1923), Plbg. and Htg., 600 Manor Rd., West New Brighton, S. I., N. Y.
PHELPS, Harold R. (*M* 1932; *A* 1932; *J* 1927), 16262 Appoline St., Detroit, Mich.
PHILIP, William (*M* 1930), 74 Bastedo Ave., Toronto, Ont., Canada.
PHILLIPS, Frederic W., Jr. (*M* 1921), 825 East 38th St., Brooklyn, N. Y.
PHILLIPS, William J. (*A* 1929), (for mail), 520 W. First St., and 1103 Forestdale Rd., Royal Oak, Mich.
PHIPPS, Frederick George (*M* 1930), Htg. Engr., Engrg. Equipment Co., Suite 420, New Birks Bldg., and (for mail), 2054 Mercier Ave., Montreal, P. Q., Canada.
PIERCE, Edgar D. (*J* 1933), Engr., Carrier Engrg. Corp., 748 E. Washington, and (for mail), 360 West 68th St., Los Angeles, Calif.
PIERCE, William MacL. (*S* 1933), 31 Potter St., Melrose, Mass.
PIHLMAN, Arthur A. (*M* 1928), (for mail), Consolidated Gas Co. of New York, 4 Irving Pl., New York, N. Y., and 235 Dwight St., Jersey City, N. J.
PILLEN, Harry A. (*A* 1933), Mfg. Repr., Htg. & Power Equipment (for mail), 622 Broadway, and 2208 Crane Ave., Cincinnati, Ohio.
PINDER, Percy H. (*M* 1919), 366 Third Ave., New York, N. Y.
PINES, Sidney (*M* 1920), Vice-Pres. (for mail), Natkin & Co., 2020 Wyandotte St., and 5225 Charlotte St., Kansas City, Mo.
PISON, Donato, Jr. (*S* 1933), c/o Philippine Natl. Bk., New York, N. Y.
PITCHER, Lester J. (*M* 1929; *A* 1928; *J* 1924), 8129 Dante Ave., Chicago, Ill.
PITTOCK, Louis B. (*M* 1930), (for mail), 429-B Oliver Bldg., and 80 Berry St., Crafton Sta., Pittsburgh, Pa.
PLACE, Clyde R. (*M* 1924), Consulting Engr. (for mail), 420 Lexington Ave., and 333 East 57th St., New York, N. Y.
PLAENERT, Alfred B. (*A* 1933; *J* 1927), 1102 S. Park St., Madison, Wis.

PLASS, Charles Webster (*M* 1928), 826 E. Haines St., Philadelphia, Pa.
PLAYFAIR, George Alexander (*A* 1924), Mgr. (for mail), Johnson Temperature Regulating Co. of Canada, Ltd., 97 Jarvis St., Toronto, and West Hill, Ont., Canada.
PLEWES, Stanley E. (*M* 1917), Philadelphia Mgr. (for mail), Johnson Service Co., 2853 North 12th St., North Philadelphia, Sta. 8., Philadelphia, and 309 Evergreen Rd., Jenkintown, Pa.
PLUNKETT, John H. (*M* 1925), Chief of Inspections, Dept. of Public Safety, 3 Hancock St., and (for mail), 81 Woodrow Ave., Boston, Mass.
POEHNER, Robert E. (*M* 1928), Vice-Pres.-Secy., W. H. Johnson & Son Co., 330 E. St. Joe St., and (for mail), 2308 Coyner Ave., Indianapolis, Ind.
POGALIES, Louis H. (*M* 1931), Mech. Engr., 4614 Prospect Ave., and (for mail), 4102 Archwood Ave., Cleveland, Ohio.
POHLE, Kenneth F. (*A* 1930), Vice-Pres., W. F. Hirschman Co., Inc., 205 East 42nd St., New York, N. Y.
POLDERMAN, Lambert H. (*M* 1927), Vice-Pres. (for mail), Carrier Engrg. Corp. of Calif., 748 E. Washington Blvd., and 1330 Colorado Blvd., Eagle Rock, Los Angeles, Calif.
POLLARD, Alfred L. (*A* 1932), Gen. Supt., Steam Heat Dept. (for mail), Puget Sound Power & Light Co., 601 Electric Bldg., and 3009 28th W. Seattle, Wash.
POPE, S. Austin (*M* 1917), Pres. (for mail), William A. Pope Co., 26 N. Jefferson St., Chicago, and 831 Ashland Ave., River Forest, Ill.
PORTER, Herbert M. (*M* 1931), 65 North 17th St., Minneapolis, Minn.
PORTER, Richmond Clay (*A* 1931; *J* 1930), College of Engrg., University of Kentucky, Lexington, Ky.
POSEY, James (*M* 1919), Consulting Engr., 4005 Liberty Heights Ave., Baltimore, Md.
POWELL, Knox A. (*S* 1933), 1008-18th Ave. S.E., Minneapolis, Minn.
POWERS, Fred I. (*M* 1920), Factory Repr. (for mail), Box 324, and 605 S. Sixth Ave., Bozeman, Mont.
POWERS, Fred W. (*M* 1911), Pres. and Gen. Mgr. (for mail), The Powers Regulator Co., 2720 Greenview Ave., and 900 Castlewood Ter., Chicago, Ill.
POWERS, Lowell G. (*J* 1930), Air Cond. (for mail), 1501 Carew Tower, Cincinnati, Ohio, and 325 W. Diamond Ave., Hazelton, Pa.
PRENTICE, Oliver J. (*A* 1927), (for mail), C. A. Dunham Co., 450 E. Ohio St., and 850 Lake Shore Dr., Chicago, Ill.
PRESDEE, Cliff W. (*A* 1926), Adv. Mgr., Heating and Ventilating, 148 Lafayette St., New York, N. Y.
PRICE, Charles E. (*A* 1933), Treas. (for mail), Keeney Publishing Co., 1900 Prairie Ave., Chicago, and 1151 Chatfield Rd., Winnetka, Ill.
PRICE, Ernest H. (*S* 1932), c/o Haff Supply, Inc., Box 328, Riverhead, L. I., N. Y.
PRYIBIL, Paul L. (*A* 1932), Partner (for mail), Hucker-Pryibil Co., 1700 Walnut St., and 328 E. Phil Ellena St., Philadelphia, Pa.
PRYOR, Frederick L. (*M* 1913), 5 Colt St., Paterson, N. J.
PURCELL, Arthur J. (*M* 1914), 631 New Britain Ave., Hartford, Conn.
PURCELL, Frederick C. (*M* 1926), Dist. Mgr. (for mail), National Regulator Co., 2847 Grand River Ave., and 18680 Santa Rosa Dr., Detroit, Mich.
PURCELL, Robert E. (*M* 1916), Htg., Plbg. and Vtg. (for mail), 1735 Willis Ave. W., and 4061 Seebaldt Ave., Detroit, Mich.
PURDY, A. K. (*M* 1922), Pres. (for mail), Purdy Mansell, Ltd., 63 Albert St., and 30 Glenrose Ave., Toronto, Ont., Canada.
PURDY, Randall B. (*A* 1927), Assoc. Editor Power (for mail), McGraw-Hill Publishing Co., 330 West 42nd St., New York, and 224-05-139th Ave., Laurelton, L. I., N. Y.
PURINTON, Dexter J. (*A* 1923), Associate (for mail), Voorhees, Gmelin & Walker, Archts., 101 Park Ave., New York, N. Y., and 23 Sachem Rd., Greenwich, Conn.
PURSELL, H. E. (*M* 1919), Special Repr., Kewanee Boiler Corp., Kewanee, Ill.
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QUINLIVAN, Laurence P. (*S* 1933), Asst., Case School of Applied Science, and (for mail), 13711 Earlewood Rd., Cleveland, Ohio.
QUIGLEY, William J. (*M* 1920), 27 Knowlton Ave., Kenmore, N. Y.
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RAINE, John J. (*M* 1912), Vice-Pres. (for mail), The G. S. Blodgett Co., 190 Bank St., and Essex Jct., V.P., Burlington, Vt.
RAINGER, Wallace F. (*A* 1930; *J* 1924), 441 Hawthorne Ave., Yonkers, N. Y.
RAISLER, Robert K. (*A* 1933; *J* 1930), Treas. (for mail), Raisler Htg. Co., 129 Amsterdam Ave., and 25 East 77th St., New York, N. Y.
RAMSEY, Raymond F. (*S* 1933), 1522 Coutant, Lakewood, Ohio.
RANCK, Guy L. (*A* 1933), Office Mgr., C. A. Dunham Co., 3605 Laclede Ave., St. Louis, and (for mail), 472 Pasadena Ave., Webster Groves, Mo.
RANDALL, W. Clifton* (*M* 1928), Detroit Steel Products Co., 2250 E. Grand Blvd., Detroit, Mich.
RANDOLPH, Charles H. (*M* 1930; *A* 1928; *J* 1926), Air Cond. Engr., The Milwaukee Electric Railway & Light Co., 231 W. Michigan St., and (for mail), 1925 N. Prospect Ave., Milwaukee, Wis.
RASMUSSEN, Robert P. (*M* 1931), Pres., Air Conditioning Economy Equipment Co., 6835 Wentworth Ave., and (for mail), 1243 East 46th St., Chicago, Ill.
RATHBUN, Perry W. (*M* 1933), Inspector, U. S. War Dept., Ft. Sill, and (for mail), 301 Ft. Sill Blvd., Lawton, Okla.
RATHER, Max F. (*M* 1919), Johnson Service Co., 2142 East 19th St., Cleveland, Ohio.
RAY, Lewis B. (*M* 1932), Pres. (for mail), Ray Engrg. Co., Inc., 800 Broad St., Newark, and 151 Augusta St., Irvington, N. J.
RAYMOND, Fred I.* (*A* 1929), Pres. (for mail), F. I. Raymond Co., 629 W. Washington Blvd., Chicago, and River Forest, Ill.
REARDON, John A. (*M* 1921), Reardon Bros. Co., 341 Union St., Lynn, Mass.
RECK, William E. (*M* 1927), 15 Esromgade, Copenhagen, Denmark.
REDFIELD, Clarke (*S* 1932), 318 Engle St., Tenafly, N. J.

REDSTONE, Arthur L. (*M* 1931), Research Engr. (for mail), Proctor & Schwartz, Inc., Seventh St. and Tabor Rd., and Park Towers, Kemble and Ogontz Ave., Philadelphia, Pa.

REED, John F. (*M* 1927; *A* 1923), Vice-Pres. (for mail), American Air Filter Co., 420 Lexington Ave., New York, and 67 Sagmore Rd., Bronxville, N. Y.

REED, Paul L. (*A* 1932), Assoc. Editor, "Air Conditioning," 167 Madison Ave., New York, N. Y., and (for mail), 1030 Childress Ave., St. Louis, Mo.

REED, Van Applegate, Jr. (*M* 1930), Federal Engrg. Co., 239 Fourth Ave., Pittsburgh, Pa.

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REESE, Philip R. (*M* 1931), Mgr. (for mail), Herman Nelson Corp., 410 Essex Bldg., and Maryland Hotel, Minneapolis, Minn.

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REID, Herbert F. (*A* 1932), Reid-Graff Plbg. Co., 1417 Peck St., Muskegon Heights, Mich.

REILLY, Charles E. (*J* 1928), 4920 City Line Ave., Philadelphia, Pa.

REILLY, J. Harry (*M* 1931; *A* 1931; *J* 1929), 14 Watson Ave., East Orange, N. J.

REINKE, Alfred G. (*J* 1933), Group Leader on Relays, Westinghouse Electric & Mfg. Co., 95 Orange St., Newark, and (for mail), 319 Park Pl., Irvington, N. J.

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RENTE, Harry W. (*M* 1931), Htg. Engr., Oil Burners, 70 W. Chippewa St., and (for mail), 114 Morris Ave., Buffalo, N. Y.

RENTE, Sidney R. (*A* 1930), 18 Charleston Ave., Kenmore, N. Y.

RETTEW, Harvey F. (*M* 1929), Htg. and Vtg. Engr., Bd. of Education, 21st and Parkway, and (for mail), 6821 Martins Mill Rd., Philadelphia, Pa.

REUSS, Edward H., Jr. (*M* 1921; *A* 1919), 49th and Grays Ave., Philadelphia, Pa.

REYNOLDS, Jack A. (*S* 1933), Box 1230, College Sta., Texas.

REYNOLDS, Thurlow W. (*M* 1922), Consulting Engr., 100 Pinecrest Dr., Hastings-on-Hudson, N. Y.

REYNOLDS, Walter V. (*A* 1928), Janes-Reynolds Co., Inc., 218 East 52nd St., New York, N. Y.

RHEA, Chester A. (*A* 1931), Steel Boiler Repr., National Radiator Corp., and (for mail), 722 Carpenter Lane, Philadelphia, Pa.

RICE, C. J. (*A* 1923), Pres. (for mail), Sterling Engrg. Co., 3738 N. Holton St., and 3376 N. Summit Ave., Milwaukee, Wis.

RICHARD, Edwin J. (*M* 1933), Owner (for mail), Chamber of Commerce Bldg., and 3564 Paxton Rd., Cincinnati, Ohio.

RICHARD, I. T. (*A* 1928), Steamfitter and Engr., James G. Gillespie Co., 708 Columbia Rd., and (for mail), 29 Virginia St., Dorchester, Mass.

RICHARDSON, Henry Thomas (*A* 1930), Vice-Pres. (for mail), Richardson & Boynton Co., 244 Madison Ave., and 156 East 79th St., New York, N. Y.

RICHMOND, John (*S* 1933), 5035 Forbes St., Pittsburgh, Pa.

RICHTMANN, William M.* (*A* 1932; *J* 1926), Instructor in Mech. Engrg. (for mail), Texas College of Arts & Industries, and 729 W. Alice St., Kingsville, Texas.

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RIESMEYER, Edward H., Jr. (*J* 1930), Htg. Engr., Schaffer Htg. Co., 231-33 Water St., and (for mail), 4702 Stanton Ave., Pittsburgh, Pa.

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RILEY, Champlain L. (*M* 1906), (*Presidential Member*), (Pres., 1921; 1st Vice-Pres., 1920; Council, 1918-1922), Clark-MacMullen & Riley, Inc., 101 Park Ave., New York, N. Y.

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RINEHART, Wilson R. (*J* 1932), Research and Elec. Engr., W. F. Hirschman Co., and (for mail), 5 Craigie St., Le Roy, N. Y.

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RITCHIE, Edmund J. (*M* 1923), Vice-Pres., Sales (for mail), Sarco Co., Inc., 183 Madison Ave., New York, and 19 Grace Court, Brooklyn, N. Y.

RITCHIE, William (*M* 1909) Vice-Pres., Boynton Furnace Co., 373 Fourth Ave., New York, N. Y., and (for mail), 17 Van Reipen Ave., Jersey City, N. J.

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ROBERTS, Henry L. (*M* 1916), Engr. and Contractor, 228 North 10th St., Philadelphia, and (for mail), 1014 Allston Rd., Brookline, Delaware Co., Pa.

ROBINSON, Harry C. (*M* 1930), Htg. Engr., 676 Pleasant St., Worcester, Mass.

ROCKWELL, Theodore F. (*M* 1933; *A* 1933; *J* 1932), Instructor (for mail), Carnegie Institute of Technology, Schenley Park, and 5721 Stanton Ave, E.E., Pittsburgh, Pa.

RODENHEISER, George B. (*M* 1933), Head Htg. and Vtg. Dept. (for mail), David Ranken, Jr., School of Mech. Trades, 4421 Finney Ave., and 3639 A, Dover Pl., St. Louis, Mo.

RODMAN, Robert W. (*M* 1922), Supt. of Plant Operation (for mail), Bd. of Education, City of New York, 500 Park Ave., and 175 West 73rd St., New York, N. Y.

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ROOKS, Alfred W. (*M* 1931), 146 Santa Ana Ave., Long Beach, Calif.

ROSE, William K. (*M* 1930), Tall Timbers, North Park, Gerrards Cross, Buckinghamshire, England.

ROSEBERRY, John H. (*M* 1931), 32 Wardman Rd., Kenmore, N. Y.

ROSEBROUGH, Robert M. (*M* 1920), Br. Mgr. (for mail), L. J. Mueller Furnace Co., 4246 Forest Park Blvd., and 6012 McPherson Ave., St. Louis, Mo.

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ROSKE, Frederick M. (*J* 1930), Secy., Automatic Combustion Engrs., Inc., of New Jersey, 17 DeNormandie Ave., Fair Haven, and (for mail), 22 Mason Pl., Keansburg, N. J.

ROSS, John O.* (*M* 1920), Ross Industries Corp., 122 East 42nd St., New York, N. Y.

ROTH, Charles F. (*A* 1930), Mgr., International Htg. and Vtg. Exposition, Grand Central Palace, New York, and (for mail), 141 East 36th St., New York, N. Y. (Nov. 1 to April 30), and Dreamthorp, Bedford Village, N. Y. (May 1 to Oct. 31).
ROTHROCK, John T. (*M* 1920), 311 Manor Rd., Douglaston, L. I., N. Y.
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ROTZ, John M. (*M* 1918), Owner (for mail), J. M. Rotz Engrg. Co., 817 Merchants Bk. Bldg., Indianapolis, and R. R. No. 2, Carmel, Ind.
ROWE, William A. (*M* 1921), (Council, 1929-1931), 718 Longfellow Ave., Detroit, Mich.
ROWLEY, Frank B.* (*M* 1918), (*Presidential Member*), (Pres., 1932; 1st Vice-Pres., 1931; 2nd Vice-Pres., 1930; Council, 1927-1933), Prof. of Mech. Engrg. and Director of Experimental Engrg. Lab., University of Minnesota, and (for mail), 4801 E. Lake Harriet Blvd., Minneapolis, Minn.
ROYER, Earl B. (*M* 1928), Designing Engr., Fosdick & Hilmer, Consulting Engrs., 1703 Union Trust Bldg., and (for mail), 6635 Iris Ave., Cincinnati, Ohio.
ROZETT, William, Jr. (*S* 1932), 3528 E. Tremont Ave., New York, N. Y.
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RUFF, DeWitt C. (*M* 1922), Healy-Ruff Co., 765 Hampden Ave., St. Paul, Minn.
RUGART, Karl (*A* 1924), Dist. Mgr. (for mail), Warren Webster & Co., 158 North 20th St., Philadelphia, and 5830 Willows Ave., W. Philadelphia, Pa.
RUPPERT, Edward H. (*A* 1923), 85 Eastern Pkwy., Brooklyn, N. Y.
RUSSELL, Joseph Nelson (*M* 1899), Managing Dir. (for mail), Rosser & Russell, Ltd., Romney House, Marsham St., Westminster and Dartmouth House, Gerrards Cross, Buckinghamshire, England.
RUSSELL, W. A. (*M* 1921), Mgr., K. C. Br. (for mail), U. S. Radiator Corp., 1405 West 11th St., and 1440 West 61st Ter., Kansas City, Mo.
RUSSELL, William B. (*M* 1928), Colorado Ave., R. F. D. No. 1., Joliet, Ill.
RYAN, Harry J. (*M* 1922), 47 Harris Ave., Albany, N. Y.
RYAN, John E. (*M* 1931), 660 Locust St., Mt. Vernon, N. Y.
RYAN, William F. (*J* 1933), Sale Engr., Lee Hardware Co., and (for mail), 609 East Iron-Salina, Kan.
RYDELL, Carl A. (*M* 1931; *A* 1931; *J* 1928), Co-partner (for mail), Clarid Engrg. Co., 168 Dartmouth St., Boston, and 280 Quinobequin Rd., Waban, Mass.

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SADLER, C. Boone (*M* 1928), Design Draftsman (for mail), Public Works Office, 11th Naval District, and 4440 Point Loma Ave., San Diego, Calif.
SAITO, Shozo (*M* 1923), Marunouchi Bldg., Opposite Tokyo Sta., Tokyo, Japan.
SAKOUTA, Mathieu L. (*M* 1924), Consulting Engr. and Expert, Gavan, Simanskaia 4-A, Leningrad, U. S. S. R.
SALISBURY, William C. (*M* 1930), Hinckley Elec. Co., Osterville, Mass.
SANBERN, E. Nute* (*M* 1923), Engr. (for mail), Mensing & Co., 12 South 12th St., and 119 S. Haviland Ave., Philadelphia, Pa.
SANDS, Olive C. (*M* 1929), G. P. O. Box 601 F. F., Sydney N. S. W., Australia.
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SANFORD, Arthur L. (*M* 1915), Space Htg. Engr. (for mail), Minneapolis-Honeywell Co., 43 East Ohio St., and 2302 Fremont Ave. S., Minneapolis, Minn.
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SAUNDERS, Laurance P. (*M* 1933), Dir. of Engrg., Harrison Radiator Corp., Lockport, N. Y.
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SCHANZE, A. Gale (*A* 1925), Sales Engr., International Engrg. Works, Inc., Framingham, and (for mail), 71 Gilbert Rd., Belmont, Mass.
SCHEIDECKER, Daniel B. (*A* 1919), Secy. (for mail), Hunter-Clark Vtg. System Co., 2800 Cottage Grove Ave., and 4626 N. Kilbourn Ave., Chicago, Ill.
SCHERNBECK, Fred H. (*A* 1930), Salesman (for mail), Wm. Bros., Boiler & Mfg. Co., Nicollet Island, and 5045 Portland Ave., Minneapolis, Minn.
SCHLEYER, Emil F. (*A* 1931), 1725 East 37th St., Brooklyn, N. Y.
SCHLICTING, Walter G. (*M* 1932), Mgr. Air Cond. Dept., Clarage Fan Co., and (for mail), 528 Pinehurst Blvd., Kalamazoo, Mich.
SCHLOSSMAN, Melvin B. (*S* 1933), c/o L. Farbman, 601 West 160th St., New York, N. Y.
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SCHNEIDER, William G. (*M* 1932), (for mail), Copper and Brass Research Assn., 25 Broadway, Room 1326, New York, and 45 Wayne Ave., White Plains, N. Y.
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SCHOENOFF, Alfred E. (*S* 1930), Schoenoff Plbg. & Htg. Co., 515 E. Second St., Menomonie, Wis.
SCHOEPFLIN, Paul H. (*M* 1920), Niagara Blower Co., 6 East 45th St., New York, N. Y.
SCHRADER, Charles C.* (*A* 1925; *J* 1923), Armstrong Cork Co., Research Div., Lancaster, Pa.
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SCHULTE, Ross R. (*M* 1931), Consulting Engr. (for mail), 1000 Guardian Bldg., and 1781 Hague Ave., St. Paul, Minn.
SCHULZ, Howard I. (*A* 1915), Crane Co., 1223 W. Broad St., Richmond, Va.
SCHULZE, Benjamin H. (*M* 1921), Dist. Mgr. (for mail), Kewanee Boiler Corp., 37 West 39th St., and 7 Park Ave., New York, N. Y.
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SCHWEIKERT, John N. (*M* 1931), 17825 St. Clair Ave., Cleveland, Ohio.
SCHWEIM, Henry J. (*M* 1928), Chief Engr. and Secy. (for mail), Gypsum Assn., 211 W. Wacker Dr., and 7360 N. Hoyne Ave., Chicago, Ill.
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SCOTT, George M. (*M* 1915), Vice-Pres. (for mail), Child & Scott-Donohue, Inc., 112 Wooster St., New York, N. Y., and 66 Bowman Ave., Port Chester, N. Y.
SCOTT, William P., Jr. (*J* 1933), 9 Scenic Way, San Francisco, Calif.
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SEELIG, Alfred E. (*M* 1926), Pres. and Gen. Mgr., L. J. Wing Mfg. Co., 154 West 14th St., and (for mail), 310 Convent Ave., New York, N. Y.
SEELIG, Lester (*M* 1925), Mech. Engr., Museum of Science & Industry, Jackson Park, and (for mail), 725 Irving Park Blvd., Chicago, Ill.
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SELLMAN, Nils T. (*M* 1922), Dir. of Sales and Utilization and Asst. Secy. (for mail), Consolidated Gas Co. of New York, 4 Irving Pl., New York, and 56 Walworth Ave., Scarsdale, N. Y.
SENIOR, Richard L. (*M* 1925), (for mail), R. L. Senior, Inc., 103 Park Ave., New York, and 10 Cherry Ave., New Rochelle, N. Y.
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SHANKLIN, John A. (*M* 1928), Secy-Treas. (for mail), West Virginia Htg. & Plbg. Co., 233 Hale St., and 1507 Quarrier St., Charleston, Va.
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SHAW, Edgar (*M* 1923), Pres. (for mail), Lynch & Woodward, Inc., 320 Dover St., Boston, and 51 Royal St., Quincy, Mass.
SHAW, Norman J. H. (*M* 1927; *J* 1925), 37 Benjamin Rd., Arlington, Mass.
SHAWLIN, Walter C. (*A* 1931), 696 S. Oak Park Court, Milwaukee, Wis.
SHEA, Michael B. (*M* 1921), Sales Dept. (for mail), American Radiator Co., 1344 Broadway, Detroit, and 117 Massachusetts Ave., Highland Park, Mich.
SHEARS, Matthew W. (*M* 1922), 39 Sylvan Ave., Toronto, Ont., Canada.
SHEFFIELD, Edward B. (*M* 1921), Armstrong Cork & Insulation Co., 522 King St., W. Toronto, Ont., Canada.
SHEFFLER, Morris (*M* 1921), Pres. (for mail), Sheffler-Gross Co., 203 Drexel Bldg., and 5451 Lebanon Ave., Philadelphia, Pa.
SHELDON, Nelson E. (*M* 1927), Dist. Sales Mgr. (for mail), Carrier Products Corp., 916 Temple Bldg., and 942 Genesee Park Blvd., Rochester, N. Y.
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SHEPPARD, William G. F. (*M* 1922), Partner (for mail), Sheppard & Abbott, 119 Harbord St., and 1 Clarendon Ave., Toronto, Ont., Canada.
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SHERMAN, Ralph A. (*M* 1933), Fuel Engr. (for mail), Battelle Memorial Institute, 505 King Ave., and 1893 Coventry Rd., Columbus, Ohio.
SHERRY, Raymond W. (*M* 1927), 601 Peace St., Hazelton, Pa.
SHIPP, Clarence C. (*M* 1923), C. C. Shipp & Co., Room 210, 230 E. Ohio St., Indianapolis, Ind.
SHIVERS, Paul F. (*M* 1930), Minneapolis-Honeywell Regulator Co., W. Canal St., Wabash, Ind.
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SHORB, Will A. (*M* 1909), Treas., The Field & Shorb Co., 709 N. Pine St., and (for mail), 3 Lincoln Pl., Decatur, Ill.
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SHUFELT, Howard M. (*A* 1928), 10245 Charles St., Chicago, Ill.
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SIEBS, Claude T. (*A* 1927), Service Systems Engr. (for mail), Western Electric Co., Inc., 11 Park Pl., New York, N. Y., and Russell Rd., Fanwood, N. J.
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SKELLY, John F. (*M* 1921), 314 Ford St., Ogdensburg, N. Y.
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SKINNER, Henry W. (*M* 1920), Consulting Engr. (for mail), Box 1334, and 4816 Dexter, Fort Worth, Texas.
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SMITH, Harold P. (*M* 1928), Pres., Automatic Combustion Engrs., Inc. of New Jersey, 17 DeNormandie Ave., Fair Haven, N. J.
SMITH, Jared A. (*A* 1933), Br. Mgr. (for mail), The Bryant Heater Co., 626 Broadway, and 3817 Indian View Ave., Mariemont, Cincinnati, Ohio.
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SMITH, Milton S. (*M* 1919), Vice-Pres., Carrier Corp., 850 Frelinghuysen Ave., Newark, and (for mail), 13 N. Terrace, Maplewood, N. J.
SMITH, Philip C., Jr. (*M* 1928), Northport, L. I., N. Y.
SMITH, Robert Hugh (*S* 1933), 4921 Forbes St., Pittsburgh, Pa.
SMITH, Wilbur F. (*M* 1920), 422 Bryn Mawr Ave., Cynwyd, Pa.
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SNELL, Ernest (*M* 1920), 3914 LeMay Ave., Detroit, Mich.
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SNYDER, Joseph S. (*A* 1925), Sales Repr., Detroit Lubricator Co., 374 Delaware Ave., and (for mail), 9 Knowlton Ave., Buffalo, N. Y.
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SODEMANN, William C. B. (*M* 1919), Pres. (for mail), Sodemann Heat & Power Co., 2306 Delmar Blvd., St. Louis, Mo.
SONNEBORN, Charles (*M* 1930), Vice-Pres. in charge of Production, Shaw-Perkins Mfg. Co., West Pittsburgh, and (for mail), R. D. No. 3, New Castle, Pa.
SOULE, Lawrence C.* (*M* 1908), Secy. and Chief Engr. (for mail), Aerofin Corp., 850 Frelinghuysen Ave., Newark, and Essex Fells, N. J.
SPAFFORD, Allen (*A* 1927), Factory Supt., Wood Conversion Co., Cloquet, Minn.
SPAFFORD, Lewis B. (*A* 1929), Consulting Engr. (for mail), Box 145, and 66 River St., Buchanan, Mich.
SPECKMAN, Charles H. (*M* 1918), Prof. Engr., 375 Bourse Bldg., Philadelphia, Pa.
SPELLER, Frank N.* (*M* 1908), Dir., Dept. of Metallurgy and Research (for mail), National Tube Co., 1922 Frick Bldg., and 6411 Darlington Rd., Pittsburgh, Pa.
SPIELMAN, Gordon P. (*A* 1931; *J* 1923), Harrison-Spielman Co., 480 Milwaukee Ave., Chicago, Ill.
SPIELMANN, Harold J. (*M* 1933), Air Cond. Engr., The Vilter Mfg. Co., and (for mail), 2549 N. Lake Dr., Milwaukee, Wis.
SPITZLEY, Ray L. (*M* 1920), 1200 W. Fort St., Detroit, Mich.
SPOFFORTH, Walter (*M* 1930), Chief of Mech. Services (for mail), U. S. Penitentiary, McNeil Island, and 1850 West Blvd., Day Island, Tacoma, Wash.
SPROULL, Howard E. (*M* 1920), Div. Sales Mgr. (for mail), American Blower Corp., 10005 American Bldg., and 3588 Raymar Dr., Cincinnati, Ohio.
SPURGEON, Joseph H. (*M* 1924), Mfrs. Repr. (for mail), 5-203 General Motors Bldg., and 17215 Pennington Dr., Detroit, Mich.
STACEY, Alfred E., Jr.* (*M* 1914), Wootton Rd., Essex Fells, N. J.
STACK, Frank Charles (*S* 1933), 140-23 Cherry Ave., Flushing, N. Y.
STACY, Stanley C. (*M* 1931), Mech. Engr. (for mail), Bd. of Education, 13 S. Fitzhugh St., and 91 Cobbs Hill Dr., Rochester, N. Y.
STAMMER, Edward L. (*M* 1919), Supt. Htg. and Vtg., Bd. of Education, Bldg., and (for mail), 4430 Tennessee Ave., St. Louis, Mo.
STANGER, Ralph B. (*M* 1920), Mgr. (for mail), Robinson & Stanger, Empire Bldg., Pittsburgh, and Deer Creek, Church Rd., Glenshaw, Pa.
STANGLAND, B. F. (*Charter Member*), (2nd Vice-Pres., 1908, Bd. of Governors, 1905-1906-1909; Bd. of Mgrs., 1895-1899; Council, 1896-1897), Morton, N. Y.
STANNARD, James M.* (*M* 1906), Pres-Treas. (for mail), Stannard Power Equipment Co., 53 W. Jackson Blvd., Chicago, and 1402 Elinor Pl., Evanston, Ill.
STAPLES, William H. (*A* 1924), Gillis & Geoghegan, 537 W. Broadway, New York, N. Y.
STARK, W. Elliott* (*M* 1926), (Council, 1932-33), Research Engr., Bryant Heater & Mfg. Co., 17825 St. Clair Ave., Cleveland, and (for mail), 1875 Rosemont Rd., East Cleveland, Ohio.
STEELE, Alfred N. (*M* 1930), 220 S. Maple Ave., Oak Park, Ill.
STEELE, John B. (*M* 1932), Chief Engr. (for mail), Engrg. Dept., Winnipeg School Bd., Ellen and William Ave., and 215 Niagara St., Winnipeg, Man., Canada.
STEELE, Maurice G. (*M* 1929), Sales Mgr. (for mail), Revere Copper & Brass, Inc., Rome Radiation Div., and 906 N. Madison St., Rome, N. Y.
STEEN, Joseph M. (*M* 1929), Iron City Htg. Co., 843 Jacksonia St., Pittsburgh, Pa.
STEINHORST, Theodore F. (*M* 1919), Engr. and Treas., Emil Steinhorst & Sons, Inc., 612 South St., and (for mail), 1664 Brinckerhoff Ave., Utica, N. Y.
STEPHENSON, George A. (*M* 1931), Apt. 3, 121 Virgil Ave., Buffalo, N. Y.
STEPHENSON, L. A. (*M* 1917), Mgr. (for mail), Powers Regulator Co., 409 East 13th St., and 801 West 57th Ter., Kansas City, Mo.
STERN, H. Richard (*M* 1923), Johnson & Morris, Inc., 233 West 18th St., New York, N. Y.
STERNBERG, Edwin (*A* 1932; *J* 1931), Air Cond. Engr., Arctic Engrg. Co., 123 White St., and (for mail), 58 East 92nd St., New York, N. Y.
STETSON, Lawrence R. (*M* 1913), 303 Congress St., Boston, Mass.
STEVENS, Harry L. (*A* 1927; *J* 1924), (for mail), M. M. Stevens Co., 108 West Sherman, and 7 West 22nd St., Hutchinson, Kans.
STEVENS, John M. (*A* 1933), 4643 Morris St., Philadelphia, Pa.
STEVENSON, Alexander R., Jr. (*M* 1932), General Electric Co., 1 River Rd., Schenectady, N. Y.
STEVENSON, Wilbur W. (*M* 1928), Steam Htg. Engr. (for mail), Allegheny County Steam Htg. Co., 435 Sixth Ave., and 1125 Lancaster Ave., Pittsburgh, Pa.
STEWART, Charles W. (*M* 1919; *A* 1918), Hoffman Specialty Co., 3707 Chrysler Bldg., New York, N. Y.
STEWART, Duncan J. (*A* 1930), Mgr., Electric Apparatus Div. (for mail), Barber-Colman Co., and 214 Franklin Pl., Rockford, Ill.

STILL, Fred R.* (*M* 1904), (*Presidential Member*), (Pres., 1918; 2nd Vice-Pres., 1917; Council, 1916-1919), Vice-Pres. (for mail), American Blower Corp., 401 Broadway, and 1 East End Ave., New York, N. Y.
STILLER, Frederick Wilbur (*J* 1933), Estimator (for mail), F. C. Stiller & Co., 129 S. Tenth St., and 138 West 49th St., Minneapolis, Minn.
STITT, Arthur B. (*S* 1933), Group Div. Head, Sears Roebuck & Co., 210 S. Broadway, and (for mail), 260 Valentine Lane, Yonkers, N. Y.
STITT, Eugene W. (*M* 1917), Sales Engr., Gas Htg. Div. (for mail), National Radiator Corp., 221 Central Ave., and 233 Mifflin St., Johnstown, Pa.
STOCKENBERG, Ruben (*M* 1922), Johnson Service Co., 1355 W. Washington Blvd., Chicago, Ill.
STOCKWELL, William R. (*M* 1903; *J* 1901), Weil-McLain Co., Michigan City, Ind.
STOEVER, George H. (*M* 1930), 7014 Cottage St., Tacony, Philadelphia, Pa.
STONE, Eugene R. (*M* 1913), 171 Harrison Ave., Boston, Mass.
STONE, George F. (*Life Member; M* 1918), Estimator, 16 Elmwood Rd., Verona, N. J.
STRACHAN, John S. (*M* 1928), 121 Furman St., Schenectady, N. Y.
STRICKLAND, Albert W. (*A* 1929), Htg. and Vtg. Engr., Big Timber, Mont.
STROBL, John Arthur (*M* 1933), Eng. and Est., The Kirk & Blum Mfg. Co., 2838 Spring Grove Ave., Cincinnati, and (for mail), 13 Martin St., St. Bernard, Ohio.
STROCK, Clifford (*A* 1929), Associate Editor (for mail), Heating and Ventilating, 148 Lafayette St., and 150 East 182nd St., New York, N. Y.
STROUSE, Sidney B. (*M* 1921), (for mail), 429 Guarantee Trust Bldg., and 22 S. Illinois Ave., Atlantic City, N. J.
STRUNIN, Jay (*J* 1933), Supt. (for mail), 408 Second Ave., and 54 West 89th St., New York, N. Y.
SULLIVAN, Daniel A. (*M* 1923), Engr. and Contractor, 3178 Rochambeau Ave., New York, N. Y.
SUMMERS, Ernest T. (*A* 1930), Pres. (for mail), Summers-Darling & Co., 121 Smith St., and Ste., 22 Newcastle Apts., Winnipeg, Man., Canada.
SUNDELL, Samuel S. (*S* 1933), 3040 Longfellow Ave., Minneapolis, Minn.
SUTCLIFFE, Arthur G. (*M* 1922; *A* 1918), Chief Engr., Ilg Elec. Vtg. Co., 2850 N. Crawford Ave., and (for mail), 4146 N. St. Louis Ave., Chicago, Ill.
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SWANEY, Carroll R. (*M* 1929; *J* 1921), Gilbert Howe Gleason, 25 Huntington Ave., Boston, Mass.
SWANSON, Harry (*M* 1933), Engr. (for mail), The Fels Co., 42 Union St., Portland, and Box 135, Cape Cottage, Maine.
SWANSTROM, Alfred E. (*S* 1932), Construction Foreman, U. S. Dept. of Interior, and (for mail), 1444 Van Buren St., St. Paul, Minn.
SWENSON, John E. (*A* 1930), Industrial Engr. (for mail), Minneapolis Gas Light Co., 800 Hennepin Ave., and 1102 S.E. 13th Ave., Minneapolis, Minn.
SYLVAN, Stig G. (*M* 1930), 215 Central Ave., Louisville, Ky.
SYSKA, Adolph G. (*M* 1933), Consulting Engr., Syska & Hennessy, 420 Lexington Ave., New York, N. Y.
SZEKELY, Ernest (*M* 1920), Vice-Pres. and Gen. Mgr. (for mail), Bayley Blower Co., 1817 South 66th St., and 3104 W. Kilbourn Ave., Milwaukee, Wis.
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TARR, Harold M. (*M* 1931), Htg. Engr., 21 Montague St., Arlington Heights, Mass.
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TAYLOR, John H. (*A* 1928), 9344–215 Pl., Queens Village, L. I., N. Y.
TAYLOR, Kenneth A. (*J* 1930), 1375 Fremont Pl., Elizabeth, N. J.
TAZE, Donovan L. (*M* 1931), Br. Mgr. (for mail), American Blower Corp., 401 Broadway, New York, N. Y.
TEASDALE, Lawrence A. (*M* 1926), Partner (for mail), Office of Hollis French, 20 Ashmun St., and 199 Nicoll St., New Haven, Conn.
TEELING, George A. (*M* 1930), Consulting Engr., Htg. and Vtg., 12 Pine St., Albany, N. Y.
TEMPLE, Walter J. (*M* 1931), Engr., J. A. Temple & Co., 919 E. Michigan Ave., and (for mail), 1215 Reed St., Kalamazoo, Mich.
TEMPLIN, Charles L. (*M* 1921), Dist. Mgr. (for mail), Carrier Corp., Bona Allen Bldg., and 781 Sherwood Rd. N.E., Atlanta, Ga.
TEN BROOK, Charles S. (*J* 1930), 1009 W. Fourth St., Duluth, Minn.
TENKONOHY, Rudolph J. (*M* 1923), Vice-Pres. (for mail), Airtherm Mfg. Co., 1474 S. Vandeventer, St. Louis, Mo., and Route 1, Dearborn, Mich.
TENNANT, Raymond J. J. (*A* 1929), Supervisor of Sales (for mail), Duquesne Light Co., 435 Sixth Ave., and 520 Navato Pl., Pittsburgh, Pa.
TENNEY, Dwight (*M* 1932), Pres. and Chief Engr. (for mail), Tenney Engrg., Inc., 40 Dickerson St., Newark, and 33 Summit Rd., Verona, N. J.
TETEREVNIKOFF, Nikolas N. (*M* 1930), Consulting Engr., 3 Fifth Soviet St., Leningrad, U. S. S. R.
THEORELL, Hugo G. T.* (*Life Member; M* 1902), Consulting Engr., Hugo Theorells Ingenieussbyra, 4, Skoldungagatan, Stockholm, Sweden.
THINN, Christian A.* (*M* 1921), C. A. Dunham Co., 450 E. Ohio St., Chicago, Ill.
THOMAS, Melvern F. (*M* 1909), Consulting Engr. (for mail), Thomas & Wardell, 220 College St., and 24 Rivercrest Rd., Toronto, Ont., Canada.
THOMAS, Norman A. (*M* 1928), Pres. (for mail), Thomas Htg. Co., 11th and Herrick Ave., and 824 Monroe Ave., Racine, Wis.
THOMAS, Richard H. (*M* 1920), Economy Pumping Machinery Co., 3431 West 48th Pl., Chicago, Ill.
THOMMEN, Adolph A. (*A* 1929), 3400 West 61st Pl., Chicago, Ill.
THOMPSON, Charles (*A* 1927), 720–13th St., Sacramento, Calif.
THOMPSON, Donald (*J* 1932), 1434 West 77th St., Cleveland, Ohio.
THOMPSON, Nelson S.* (*M* 1917; *J* 1897), 1615 Hobart St. N.W., Washington, D. C.
THOMPSON, Richard C. (*M* 1927), 250 Stuart St., Boston, Mass.
THOMPSON, William J. (*A* 1929), Imperial Iron Corp., Ltd., St. Catharines, Ont., Canada.
THOMSON, Thomas N.* (*M* 1899), Consulting Engr., 37 Irwin Pl., Huntington, L. I., N. Y.

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THORNTON, Frank, Jr. (*A* 1930), Engrg. Mgr., Association Activities, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa.
THORNTON, Roger T. (*M* 1919), Buffalo Forge Co., 490 Broadway, Buffalo, N. Y.
THORNTON, William B.* (*M* 1931), Sales Engr. (for mail), Carrier Engrg. Corp., Bona Allen Bldg., Atlanta, and 155 Coventry Rd., Decatur, Ga.
THRUSH, Homer A. (*M* 1918), H. A. Thrush & Co., 21-23 E. River St., Peru, Ind.
TIBBITS, John C. (*M* 1920), Engrg. Dept., B. & O. R. R. Co., and (for mail), P. O. Box 106, Ellicott City, Md.
TICHENOR, Leslie R., Jr. (*J* 1930), L. R. Tichenor & Son., 1259 Robert St., Hillside, N. J.
TILDEN, Elwyn E. (*M* 1924), Asst. Mgr. (for mail), Warren Webster & Co., 76 Summer St., Boston, and Holbrook, Mass.
TILLER, Louin (*S* 1933), 1135 Northwest 41st St., Oklahoma City, Okla.
TILTZ, Bernard E. (*M* 1930), Pres. (for mail), Tiltz Air Conditioning Corp., 480 Lexington Ave., New York, and 24 Barnum Rd., Larchmont, N. Y.
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TIMMIS, W. Walter (*M* 1923; *A* 1925), Engr. (for mail), American Radiator Co., 40 West 40th St., New York, and Oak Lane, Glen Cove, N. Y.
TISNOWER, William (*M* 1923), 131 Livingston St., Brooklyn, N. Y.
TITUS, Marvin S. (*M* 1928), 478 Lamson Ave., Bedford, Ohio.
TJERSLAND, Alf (*M* 1916; *J* 1906), E. Sunde & Co., Ltd., Oslo, Norway.
TOBIN, George J. (*M* 1905), Owner, Sanitary, Htg. and Vtg. Engr., 187 North Ave., Plainfield, N. J.
TOOKER, Charles C. (*M* 1918), Owner (for mail), Tooker Plbg. & Htg., 116 North 27th St., and 208 Terry Ave., Billings, Mont.
TOONDER, Clarence L. (*M* 1933), Air Cond. Eng., Sales Engrg. Dept., Kelvinator Corp., Plymouth Road, and (for mail), 12761 Strathmoor Ave., Detroit, Mich.
TORR, Thomas W. (*M* 1933), Rudy Furnace Co., Dowagiac, Mich.
TORRANCE, Henry (*M* 1933), Pres., 175 Christopher St., and (for mail), 112 East 17th St., New York, N. Y.
TOUTON, R. D. (*M* 1933), Tech. Dir. (for mail), Bayuk Cigars, Inc., Ninth and Columbia Ave., Philadelphia, and 19 Lodges Lane, Cynwyd, Pa.
TOWER, Elwood S. (*M* 1930), Consulting Engr., 1411 Wightman St., Pittsburgh, Pa.
TRANE, Reuben N.* (*M* 1915), The Trane Co., and (for mail), 126 South 15th St., LaCrosse, Wis.
TRAUGOTT, Mortimer (*A* 1930), East Sales Mgr. (for mail), Bryant Heater & Mfg. Co., 152 North 15th St., Philadelphia, and 721 Meeting House Rd., Elkins Park, Pa.
TREADWAY, Quentin (*J* 1932), Sales Engr. (for mail), Clarage Fan Co., 707 Security Bank Bldg., and 2618 Collingwood, Toledo, Ohio.
TRIMMER, Charles M. (*S* 1933), Meter Testor, Meter Dept., Rockland L. & P. Co., Middletown, and (for mail), 28 Prospect St., Port Jervis, N. Y.
TROSKE, Joseph J. (*A* 1931), Vice-Pres. and Gen. Mgr. (for mail), Vander-Troske Co., 236 Winter Ave. N.W., and 233 Brown St. S.E., Grand Rapids, Mich.
TRUITT, Joseph E. (*M* 1920; *A* 1911), Pres., Autovent Fan & Blower Co., 1805 N. Kostner Ave., Chicago, Ill.
TRULSON, Arthur F. (*M* 1930), Mech. Engr., 1509 W. Sixth St., Ashland, Wis.
TRUMBO, Silas M. (*A* 1926), Sales (for mail), Buffalo Forge Co., 15 N. Jefferson St., Chicago, and 921 Franklin St., Downers Grove, Ill.
TUCKER, Frank N. (*M* 1926), Field Engr., Ilg Electric Vent. Co., Room 1108, 13 Park Row, New York, and (for mail), 239 Whaley St., Freeport, L. I., N. Y.
TUCKERMAN, George E. (*M* 1932), 6202 Ogontz Ave., Philadelphia, Pa.
TULLOSS, Joseph C. (*M* 1930), 814 Connecticut Ave., N.W., Washington, D. C.
TURLAND, Charles H. (*A* 1930), Mgr., Htg. and Vtg. Dept., Kipp-Kelly, Ltd., 68 Higgins Ave., and (for mail), 325 Centennial St., Winnipeg, Man., Canada.
TURNAU, Edmund H. (*S* 1933), Cadet Engr., Koppers Seaboard By-Product Coke Co., and (for mail), 23 Polifly Rd., Hackensack, N. J.
TURNER, John (*M* 1930), 115 Myrtle St., Boston, Mass.
TURNER, John W. (*M* 1928), 26031 Concord Rd., Royal Oak, Mich.
TURNO, Walter G. W. (*M* 1917; *A* 1912), Secy., H. W. Porter & Co., Newark, and (for mail), 71 Lafayette Ave., East Orange, N. J.
TUSCH, Walter (*M* 1917), H. & V. Engr., Tenney & Ohmes, Inc., 101 Park Ave., New York, and (for mail), 881 Sterling Pl., Brooklyn, N. Y.
TUTTLE, J. Frank (*M* 1913), Mgr., Warren Webster Co., 76 Summer St., Boston, and 2 Elmwood Ave., Winchester, Mass.
TUVE, George L. (*M* 1932), Asso. Prof. of Mech. Engrg. (for mail), Case School of Applied Science, and 1294 Cleveland Heights Blvd., Cleveland, Ohio.
TWIST, Charles F. (*M* 1921), Secy. (for mail), Ashwell-Twist Co., 967 Thomas St., and 2310 Tenth Ave. N., Seattle, Wash.
TYLER, Roy D. (*M* 1928), East Sales Mgr. (for mail), Modine Mfg. Co., 101 Park Ave., New York, and 15 Highbrook Ave., Pelham, N. Y.
TYSON, William H. (*M* 1928), Mgr. of Engrg. (for mail), Goodyear Tyre & Rubber Co., Ltd., and "Kipewa" Codsall Rd. N.R., Wolverhampton, England.

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ULLMAN, Herbert G. (*A* 1928), Secy. of Lab., American Radiator Co., 675 Bronx River Rd., Yonkers, and (for mail), 107 White Rd., Scarsdale, N. Y.
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VAN COURT, Walter Gearing (*M* 1930), Htg. Engr., 557 Communipaw Ave., Jersey City, N. J.
VANDERHOOF, Austin L. (*A* 1933), Northern Ohio Mgr., Kewanee Boiler Corp., and (for mail), Warren Webster & Co., 2341 Carnegie Ave., Cleveland, and 3120 Yorkshire Rd., Cleveland Heights, Ohio.

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VAN SICKLE, William B. (*M* 1915), Mgr. (for mail), W. B. Van Sickle Co., 1623 St. Clair Ave. N.E., Cleveland, and 1530 Grace Ave., Lakewood, Ohio.
VERMERE, Earl J. (*M* 1929), Sales Engr., Warren Webster & Co , 2341 Carnegie Ave., and (for mail), 2125 Wyandotte Ave., Cleveland, Ohio.
VERNIER, Marcel G. (*S* 1933), 730 Hill Ave., Wilkinsburg, Pa.
VERNON, J. Rexford (*M* 1928; *A* 1926), (for mail), Johnson Service Co., 1355 Washington Blvd., Chicago, and 1020 Austin St., Evanston, Ill.
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VINCENT, Paul J. (*M* 1931), Paul J. Vincent Co., 1010 Chandler Bldg., Washington, D. C., and (for mail), 3807 Beech Ave., Baltimore, Md.
VINSON, Neal L. (*S* 1932), (for mail), 630 Clyde St., Pittsburgh, Pa., and Box 1438, Bisbee, Ariz.
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VOGEL, Andrew (*M* 1926), Engr. (for mail), General Electric Co., and 1821 Lenox Rd., Schenectady, N. Y.
VOGELBACH, Oscar (*M* 1923), 23 William St., North Arlington, N. J.
VOGT, John H. (*A* 1925), Mech. Engr. (for mail), New York State Dept of Labor, 80 Centre St., New York, and 87 Grant Ave., Brooklyn, N. Y.
VOGT, Joseph B. (*M* 1933; *A* 1933; *J* 1929), Asst. Htg. and Vtg. Engr., New York State D. P. W., Albany, and (for mail), 1403 Park Blvd., Troy, N. Y.
VOISINET, Walter E. (*M* 1930), Sales Repr. (for mail), Herman Nelson Corp., 250 Delaware Ave., Buffalo, and 151 Warren Ave., Kenmore, N. Y.
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W

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WAECHTER, Herman P. (*A* 1930; *J* 1927), Air Cond. Engr., York Ice Machinery Corp., Brooklyn, and (for mail), 89 Sherman Ave., Tompkinsville, N. Y.
WAGNER, A. M. (*A* 1921), Mgr. (for mail), American Radiator Co., 1741 W. St. Paul Ave., and 1857 N. Prospect Ave., Milwaukee, Wis.
WAITE, Harry (*A* 1929), 1409 North 17th St., Superior, Wis.
WALDON, Charles D. (*A* 1932), 32 Ferndale Ave., Toronto, Ont., Canada.
WALKER, Alexander (*A* 1925), C. A. Dunham Co., Ltd., 1307 Fifth St. W., Calgary, Alta., Canada.
WALKER, James H.* (*M* 1916), Supt. of Central Htg. (for mail), The Detroit Edison Co., 2000 Second Ave., Detroit, and 432 Arlington Rd., Birmingham, Mich.
WALLACE, Bruce (*M* 1930), Proprietor (for mail), B. Wallace, 5 Eden St., Newmarket, Auckland S.E. 1, and 113 Western Springs Rd., Morningside, Auckland S.W., 1, New Zealand.
WALLACE, George J. (*M* 1923), Principal, 331 East 54th St., New York, and (for mail), 27-36 Ericsson St. E., Elmhurst, N. Y.
WALLACE, Kenneth S. (*M* 1931), Gas Htg. Engr., Peoples Gas Co., and (for mail), 5737 Kenmore Ave., Chicago, Ill.
WALLACE, William M., II (*M* 1929), 8908-196th St., Hollis, L. I., N. Y.
WALLICH, A. C. (*M* 1919), (for mail), Wallich Ice Machine Co., 517 E. Larned St., and 1667 Burlingame, Detroit, Mich.
WALTERS, Arthur L. (*M* 1926; *A* 1925; *J* 1924), 7284 Richmond Pl., Maplewood, Mo.
WALTERS, William T. (*M* 1917), Engr., Illinois Engrg. Co., Cor. 21st St. and Racine Ave., and (for mail), 7965 Phillips Ave., Chicago, Ill.
WALTHER, Vernon H. (*M* 1928; *J* 1925), Mech. Engr., 6821 Osceola Ave., Chicago, Ill.
WALTERTHUM, John J. (*A* 1922), Htg. Contractor, 173 East 62nd St., New York, N. Y., and (for mail), 42-A Van Reipen Ave., Jersey City, N. J.
WALTON, Hiram L.* (*M* 1916), Member of Firm (for mail), Smith-Hinchman & Grylls, 800 Marquette Bldg., Detroit, and Lake Angelus, Pontiac, Mich.
WANDLESS, Franklin W. (*M* 1925), Haynes Selling Co., Inc., 1518 Fairmount Ave., Philadelphia, and Berwyn, Pa.
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WARD, William T. (*M* 1930), 18 Castlefield Ave., Toronto, Ont., Canada.
WARING, J. M. S. (*M* 1932), Consulting Engr. (for mail), Chase & Waring, 17 East 42nd St., and 277 Park Ave., New York, N. Y.
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WARREN, Harry L. (*M* 1930), 1303 Huntington Dr., South Pasadena, Calif.
WARREN, Walter J. (*M* 1930), Engr. (for mail), 180 N. Michigan Ave., Chicago, and 118 Gillick St., Park Ridge, Ill.
WASHINGTON, Laurence W. (*M* 1929), 2301 Knox Ave., Chicago, Ill.
WATERMAN, John H. (*M* 1931), 201 Devonshire St., Boston, Mass.
WATERS, George G. (*M* 1931; *A* 1926), American Blower Co., 801 First Natl. Bk. Bldg., Pittsburgh, Pa.
WATSON, M. Barry (*M* 1928), Consulting Engr., 121 Welland Ave., Toronto 5, Canada.
WAUNG, Tsing F. (*J* 1933), Htg. Engr., Andersen Meyer & Co., and (for mail), 103 Route Remi, Shanghai, China.
WEAGER, T. A. (*M* 1920), Dist. Mgr. (for mail), Buffalo Forge Co., 418 Rockefeller Bldg., and 3124 Berkshire Rd., Cleveland, Ohio.
WEBB, John S. (*M* 1920), Sales Mgr., Webster Tallmadge Co., Inc., 300 Madison Ave., New York, N. Y., and (for mail), 16 Brookline St., Needham, Mass.
WEBB, John W. (*M* 1926), 6 Meadows Rd., Heaton Chapel, Stockport, England.
WEBSTER, E. Kessler (*M* 1915), Warren Webster & Co., 17th and Federal Sts., Camden, N. J.
WEBSTER, Warren (*Life Member* 1933; *M* 1906; *A* 1890), Pres., Warren Webster & Co., 17th and Federal Sts., Camden, N. J.
WEBSTER, Warren, Jr. (*M* 1932; *A* 1932; *J* 1927), Vice-Pres. and Treas. (for mail), Warren Webster & Co., 17th and Federal Sts., Camden, and Washington and Colonial Ridge Dr., Haddonfield, N. J.
WECHSBERG, Otto (*M* 1932), Pres. and Gen. Mgr., Coppus Engrg. Corp., 344 Park Ave., and (for mail), 12 Rosemont Rd., Worcester, Mass.
WEGMANN, Albert (*M* 1918), 6206 North 17th St., Philadelphia, Pa.
WEHRLE, Robert H. (*J* 1933), Sales Engr. (for mail), 323 E. Fourth St., and 2368 Victory Pkwy., Cincinnati, Ohio.
WEIL, Martin (*A* 1925), Vice-Pres. (for mail), Weil-McLain Co., 641 W. Lake St., and 4259 Hazel Ave., Chicago, Ill.
WEIL, Maurice I. (*A* 1928), Pres. (for mail), Chicago Pump Co., 2336 Wolfram St., and 1409 Elmdale Ave., Chicago, Ill.

WEIMER, Fred G. (*A* 1919), Local Mgr., Kewanee Boiler Corp., and (for mail), 3958 N. Stowell Ave., Milwaukee, Wis.

WEINSHANK, Theodore* (*Life Member* 1933; *M* 1906), (Bd. of Governors, 1913), 2323 N. Kedzie Blvd., Chicago, Ill.

WEISS, Arthur P. (*M* 1928), 134 Farrington Ave., North Tarrytown, N. Y.

WEISS, Carl A. (*A* 1924), Supt. (for mail), Kornbrodt Kornice Ko., 1811 Troost Ave., and 29 East 68th St., Kansas City, Mo.

WELAMB, Victor N. (*M* 1918), V. N. Welamb Co., 105 N. Watts St., Philadelphia, Pa.

WELCH, Louis A., Jr. (*A* 1929), 443 Second St., Schenectady, N. Y.

WELDY, Lloyd O. (*M* 1930), Powers Regulator Co., 2720 Greenview Ave., Chicago, Ill.

WELLS, Eric E. (*M* 1930), 1458-155th St., Beechhurst, L. I., N. Y.

WELSH, Harry S. (*M* 1906), Sales Engr. (for mail), Weil-McLain Co., 404 Atlantic Ave., and 53 Kemphurst Rd., Rochester, N. Y.

WELTER, M. A. (*A* 1925), Htg. Engr. (for mail), Welter Furnace Co., 2118 Lyndale Ave. S., and 4306 S. Garfield, Minneapolis, Minn.

WENDT, Edgar F. (*M* 1918), Pres. (for mail), Buffalo Forge Co., 490 Broadway, and 731 Lafayette Ave., Buffalo, N. Y.

WEST, Perry* (*M* 1911), (Council, 1920-1925; Treas., 1924-1925), Consulting Engr. (for mail), 13 Central Ave., and 445 Ridge St., Newark, N.J.

WHALEY, Ralph S. (*M* 1931), 1933 Fifth Ave., Seattle, Wash.

WHEELER, Charles A. (*A* 1931), Sales Engr. (for mail), Herman Nelson Corp., 400 Ninth Vincent Bldg., Cleveland, and 2121 McKinley Ave., Lakewood, Ohio.

WHEELER, Otto J. (*M* 1923), Pres-Treas. (for mail) The Samuel A. Esswein Htg. & Plbg. Co., 548-558 W. Broad St., and 2044 Collingswood Rd., Columbus, Ohio.

WHELLER, Harry S. (*M* 1916), Vice-Pres., L. J. Wing Mfg. Co., 154 West 14th St., New York, N. Y., and (for mail), 725 Union Ave., Elizabeth, N. J.

WHITE, Everett A. (*M* 1921), Engrg. Dept., Crane Co., 30 South 16th St., and (for mail), 5244 Nottingham, St. Louis, Mo.

WHITE, Elwood S. (*M* 1921), Pres. (for mail), Taco Heaters, Inc., Room 1224, 342 Madison Ave., New York, N. Y., and Meadowbank Rd., Old Greenwich, Conn.

WHITE, James J. (*A* 1930), 6035 Nassau St., Philadelphia, Pa.

WHITE, John C. (*M* 1932), State Power Plant Engr. (for mail), 624 E. Main St., and 622 E. Main St., Madison, Wis.

WHITELAW, H. Leigh (*M* 1916), Vice-Pres. (for mail), American Gas Products Corp., 40 West 40th St., New York, N. Y., and Overbrook Lane, Darien, Conn.

WHITELEY, Stockett M. (*M* 1933), Consulting Engr. (for mail), Baltimore Life Bldg., and 3931 Canterbury Rd., Baltimore, Md.

WHITTALL, Ernest T. (*A* 1933), Vice-Pres. and Managing Dir. (for mail), May Oil Burner of Canada, Ltd., 196 Adelaide St. W., and 11 Cottingham Rd., Toronto, Ont., Canada.

WHY, H. Berkeley (*M* 1919), 640 W. Sedgwick St., Philadelphia, Pa.

WIDDICOMBE, Robert A. (*M* 1903), 1120 Lake Shore Dr., Chicago, Ill.

WIEGNER, Henry B. (*M* 1919), Mgr., Boston Office, Johnson Servie Co., 20 Winchester St., Boston, and (for mail), 143 Standish Rd., Watertown, Mass.

WIERENGA, Peter O. (*A* 1931), Vice-Pres. (for mail), C. C. James Co., 49 Coldbrook St. N.E., and 231 Brown St. S.E., Grand Rapids, Mich.

WIGGINS, Oswald James (*S* 1933), Walnut Grove, Minn.

WIGGS, G. Lorne (*A* 1932; *J* 1924), Mgr., Montreal Sales Office (for mail), C. A. Dunham Co., Ltd., 608 University Tower, and 4797 Grosvenor Ave., Montreal, Que., Canada.

WIGLE, Bruce M. (*A* 1926), Pres. (for mail), Bruce Wigle Plbg. & Htg. Co., 9117 Hamilton Ave., and 18114 Oak Dr., Detroit, Mich.

WILD, Walter H. (*M* 1927; *A* 1921), Union Iron Works, Land Title Bldg., Philadelphia, Pa.

WILDER, Edward L. (*M* 1915), Mgr., Gas Sales (for mail), Utility Management Corp., 120 Wall St., New York, and 149 Mt. Joy Pl., New Rochelle, N. Y.

WILEY, Edgar C. (*M* 1909), Wiley & Wilson, Lynchburg, Va.

WILKINSON, Farley J. (*M* 1933), Engr., Montgomery Ward & Co., Chicago, and (for mail), 18257 Martin Ave., Homewood, Ill.

WILLARD, Arthur C.* (*M* 1914), (*Presidential Member*), (Pres., 1928; 1st Vice-Pres., 1927; 2nd Vice-Pres., 1926; Council, 1925-1929), Prof. Htg. and Vtg. and Head of Dept. of Mech. Engrg. (for mail), University of Illinois, and 1208 W. California St., Urbana, Ill.

WILLIAMS, Allen W. (*A* 1915), Managing Dir. (for mail), Natl. Warm Air Htg. & Air Cond. Assn., 50 W. Broad St., Columbus, and 51 Meadow Park Ave., Bexley, Ohio.

WILLIAMS, J. McFarland, Jr. (*A* 1928; *J* 1927), Sales Engr., 1407-35th St. N.W., Washington, D. C.

WILLIAMS, J. Walter (*M* 1915), Pres. (for mail), Forest City Plbg. Co., 332-36 E. State St., and 923 E. State St., Ithaca, N. Y.

WILLIAMS, Leo E. (*A* 1933; *J* 1930), 225 Arch St., Meadville, Pa.

WILLIS, Roy C. (*M* 1927), Mgr. New York Office (for mail), Vapor Engrg. Co., 597 Fifth Ave., New York, and 253 Lindell Blvd., Long Beach, N. Y.

WILMOT, Charles S. (*M* 1919), 406 Essex Ave., Narberth, Pa.

WILSON, Benjamin W. (*M* 1922), The Ballinger Co., S.E. Cor. 12th and Chestnut Sts., Philadelphia, Pa.

WILSON, George T. (*M* 1925), Tyre Ave., Islington, Ont., Canada.

WILSON, Harold A., Jr. (*J* 1933), Student salesman, American Radiator Co., 40 West 40th St., and (for mail), 1133 Park Ave., New York, N. Y.

WILSON, Harry A. (*Charter Member; Life Member*), P. O. Box 155, Washington, R. I.

WILSON, W. H. (*A* 1932), Steamfitter Foreman, Pullman Car & Mfg. Corp., 11001 Cottage Grove Ave., and (for mail), 22 West 110th Pl., Chicago, Ill.

WILSON, William H. (*A* 1923), Br. Mgr. (for mail), Johnson Service Co., 507 E. Michigan St., and 2023 E. Olive St., Milwaukee, Wis.

WINANS, Glen D. (*M* 1929), Engr. of Steam Distribution (for mail), The Detroit Edison Co., 2000 Second Ave., and 16183 Wisconsin, Detroit, Mich.

WINCH, Franklin R. (*M* 1925), Consulting Engr., Signal Oil Bldg., and (for mail), 1058 Bedford St., Los Angeles, Calif.

WINQUIST, Walter J. (*A* 1930), Htg. and Vtg. Engr., 294 Nostrand Ave., Brooklyn, N. Y.

WINSLOW, C.-E. A.* (*M* 1932), Prof. of Public Health (for mail), Yale University, 310 Cedar St., and 314 Prospect St., New Haven Conn.

WINTERBOTTOM, Ralph F. (*M* 1923), Htg. Engr., Winterbottom Supply Co., Commercial Miles, and (for mail), 1002 Riehl St., Waterloo, Iowa.

WINTERER, Frank C. (*M* 1920), Sales Mgr. (for mail), Cochran Sargent Co., Broadway and Kellogg Blvd., and 836 Juno St., St. Paul, Minn.

WINTHER, Anker (*J* 1932), Air Cond. Engr., York Ice Machinery Corp., 2116 Gilbert Ave., Cincinnati, Ohio.

WOLF, John C. (*M* 1923), (for mail), B. F. Sturtevant Co., and 265 Fairmount Ave., Hyde Park, Mass.

WOLFF, Oscar H. (*M* 1926), 6232 Oakland Ave., St. Louis, Mo.

WOLFF, Richard A. (*M* 1919; *J* 1915), Insurance also Secy. (for mail), Wolff & Munier, Inc., 500 Fifth Ave., New York, and 888 Woodmere Pl., Woodmere, L. I., N. Y.

WOOD, Frederick C. (*J* 1931), Sales Air Cond Engr. (for mail), Westerlin & Campbell Co., 1113 Cornelia Ave., and 1905 Estes Ave., Chicago, Ill.

WOOD, J. Sydney (*M* 1926), Estimator (for mail), Bennett & Wright, Ltd., 72 Queen St. E., and 50 Davisville Ave., Toronto, Ont., Canada.

WOODLING, Miner D. (*M* 1926), Owner, 1002 Greenway Ter., Kansas City, Mo.

WOOLSTON, A. H. (*M* 1919), 2015 Sansom St., Philadelphia, Pa.

WORSHAM, Herman (*M* 1925; *J* 1918), 103 N. Walnut St., E. Orange, N. J.

WRIGHT, Clarence E. (*S* 1933), (for mail), 276 N. Bellefield Ave., Pittsburgh, Pa., and 854 Maiden Lane, Roanoke, Va.

WRIGHT, Kenneth A. (*M* 1921), Mgr. (for mail), Johnson Service Co., 1113 Race St., Cincinnati, Ohio, and 113 Orchard Rd., Ft. Mitchell, Ky.

WRIGHT, M. Birney (*A* 1932; *J* 1929), Instructor (for mail), Case School of Applied Science, Cleveland, and 1056 Nela View Rd., Cleveland Heights, Ohio.

WUNDERLICH, Milton S.* (*M* 1925), Insulite Co., Minneapolis, Minn., and (for mail), 1598 Laurel Ave., St. Paul, Minn.

WYLIE, Howard M. (*M* 1925; *J* 1917), Vice-Pres. in charge of Sales (for mail), Nash Engrg. Co., and 51 Elmwood Ave., S. Norwalk, Conn.

WYMAN, Dwight M. (*M* 1931), R. F. D. 421, Barrington, R. I.

WYMORE, Fred C. (*S* 1933), 100 Westport Rd., Kansas City, Mo.

Y

YAGER, John J. (*M* 1921), 425 Woodbridge Ave., Buffalo, N. Y.

YAGLOU, Constantin P.* (*M* 1923), Asst. Prof. of Industrial Hygiene (for mail), Harvard School of Public Health, 55 Van Dyke St., Boston, and 24 Wade St., Brighton, Mass.

YARDLEY, Ralph W. (*M* 1920), Asst. Archt., Bd. of Education, City of Chicago, 228 N. LaSalle St., Room 568, and (for mail), c/o Judge J. W. Galbraith, Farmers Bk. Bldg., Suite 601, Mansfield, Ohio.

YATES, Walter (*M* 1902), Governing Dir., Matthews & Yates, Ltd., Swinton, Lancs, England.

YEAGER, George F. (*M* 1933), Inspector Test Dept., Pennsylvania Railroad, Test Dept., P. R. R., Altoona, and (for mail), 516 Crawford Ave., Altoona, Pa.

YEOMANS, Paul H. (*M* 1930), 1177 Spring Grove Ave., Lancaster, Pa.

YOCKEL, Thomas J. (*A* 1929), 1074 Franklin Ave., New York, N. Y.

Z

ZACK, Hans J. (*M* 1928), Mgr., Zack Co., 2311 W. Van Buren St., Chicago, Ill.

ZECK, Alexander (*Life Member; M* 1904), Pres. (for mail), Alex Zeck & Son Co., 902 University Ave., and 324 Willy St., Morgantown, W. Va.

ZIBOLD, Carl Edward (*M* 1929), Mech. Engr., Htg. and Vtg. Co., Colonial Ter., Westminster Ridge, White Plains, N. Y.

ZIESSE, Karl L. (*A* 1931), Secy-Treas. (for mail), Phoenix Sprinkler & Htg. Co., 115 Campau Ave. N.W., and 315 Hampton Ave. S.E., Grand Rapids, Mich.

ZIMMERMAN, Alexander H. (*A* 1930), Ventilation Engr., Chicago Bd. of Health, 707 City Hall, and (for mail), 3748 Irving Park Blvd., Chicago, Ill.

ZINK, David D. (*M* 1931), Sales Mgr., E. K. Campbell Cos., 2441 Charlotte St., Kansas City, Mo., and (for mail), C. O., 980th Co., CCC., Drain, Ore.

ZOKELT, C. G. (*M* 1921), Consulting Engr., 3810–24th Ave. S., Seattle, Wash.

ZUHLKE, William R. (*M* 1928), 530 McLean Ave., Yonkers, N. Y.

Summary of Membership

(Corrected to January 1; 1934)

UNITED STATES

Alabama	1	Minnesota	87
Arizona	1	Missouri	79
Arkansas	2	Montana	4
California	51	Nebraska	5
Colorado	4	New Hampshire	1
Connecticut	32	New Jersey	109
Delaware	5	New York	378
District of Columbia	18	North Carolina	7
Florida	2	Ohio	95
Georgia	5	Oklahoma	6
Illinois	201	Oregon	1
Indiana	24	Pennsylvania	240
Iowa	5	Rhode Island	8
Kansas	5	Tennessee	5
Kentucky	11	Texas	20
Louisiana	3	Vermont	4
Maine	3	Virginia	12
Maryland	14	Washington	22
Massachusetts	111	West Virginia	5
Michigan	112	Wisconsin	48
			1746

FOREIGN COUNTRIES

Austria	1	Italy	2
Australia	2	Japan	4
Belgium	1	Mexico	3
Canada	80	New Zealand	3
China	10	Norway	2
Czechoslovakia	1	South America	1
Denmark	1	Sweden	2
England	14	U. S. S. R.	2
France	8		
Germany	3		143
India	2		
Ireland	1	Total Membership	1889

SUMMARY OF MEMBERSHIP BY GRADES

Honorary Members	2
Presidential Members	22
Members	1244
Associate Members	364
Junior Members	132
Student Members	125
	1889

LIST OF MEMBERS
Geographically Arranged

UNITED STATES

ALABAMA

Birmingham—
Lichty, C. P.

ARIZONA

Tucson—
Moreau, D.

ARKANSAS

Ozark—
Pesterfield, C. H.

Siloam Springs—
Jones, C. R.

CALIFORNIA

Altadena—
Bremser, H. A.

Berkeley—
Duncan, G. W., Jr.

Beverly Hills—
Nelson, H. A.

Huntington Park—
Barnum, W. E., Jr.

Long Beach—
Rooks, A. W.

Los Angeles—
Barker, C. M.
Binford, W. M.
Bouey, A. J.
Bullock, H. H.
Bunker, K. S.
Cranston, W. E., Jr.
Ellingwood, E. L.
Gillespie, J. D.
Herman, J., Jr.
Hill, F. M.
Hilliard, F. H.
Holladay, W. L.
Horton, G. H.
Hungerford, L.
Keeling, H. B.
Kendall, E. H.
Kennedy, M.
Kooistra, J.
La Montagne, J. M.
Lawler, M. M.
McClelland, H. S.
Ness, W. H. C.
Newman, C. T.
Orear, A. G.
Oster, G. R.
Ott, O. W.
Park, J. F.
Pierce, E. D.
Polderman, L. H.
Scofield, P. C.
Sharpe, N.
Simonds, A. H.
Winch, F. R.

Mesa Grande—
Hemingway, W. S.

Oakland—
Cummings, G. J.

Pasadena—
Gifford, R. L.
O'Haver, H. M.

Sacramento—
Thompson, C.

San Diego—
Sadler, C. B.

San Francisco—
Corrao, J.
Haley, H. S.
Krueger, J. I.
Leland, W. E.
Scott, W. P., Jr.

South Pasadena—
Campbell, W. E.
Warren, H. L.

COLORADO

Colorado Springs—
Higgins, D. T.
Jardine, D. C.

Denver—
Daly, J. H.
Ward, O. G.

CONNECTICUT

Bridgeport—
Feydt, J. C.

Colchester—
Adams, W. H.

Fairfield—
Osborn, W. J.

Greenwich—
Jones, A. L.
Opperman, E. F.

Hartford—
Purcell, A. J.

Manchester—
Buck, L.
Millard, J. W.

New Britain—
Hjerpe, C. A., Jr.

New Haven—
Greenburg, L.
Hoyt, W. B.
Seeley, L. E.
Tavanlar, E. J.
Teasdale, L. A.
Winslow, C. E. A.

New London—
Chapin, C. G.
Forsberg, W.
Hopson, W. T.

Noroton Heights—
Ashley, E. E.

Norwalk—
Monroe, R. R.

Riverside—
Murphy, J. R.

South Norwalk—
Adams, H. E.
Harvey, A. D.
Jennings, I. C.
Lyons, C. J.
Mead, E. A.
Wylie, H. M.

Stamford—
Hoyt, L. W.

Waterbury—
Ahlberg, H. B.
Hutzel, H. F.
Simpson, W. K.

Wilton—
Humphreys, J.

DELAWARE

Wilmington—
Gawthrop, F. H.
Hayman, A. E., Jr.
Kershaw, M. G.
Lownsbery, B. F.
Schoenijahn, R. P.

DISTRICT OF COLUMBIA

Washington—
Beitzell, A. E.
Coward, H.
Dalla Valle, J. M.
Febrey, E. J.
Feltwell, R. H.
Fenn, C. V. O.
Frankel, G. S.
Gardner, S. F.
Hood, O. P.
Johnson, J. M.
Nolan, J. J., Jr.
Ourusoff, L. S.
Powers, L. G.
Smallman, E. W.
Thompson, N. S.
Tulloss, J. C.
Urdahl, T. H.
Williams, J. M., Jr.

FLORIDA

Ft. Lauderdale—
Charlton, J. F.

Palm Beach—
Hodeaux, W. L.

GEORGIA

Atlanta—
Clare, F. W.
Kent, L. F.
Klein, E. W.
Templin, C. L.
Thornton, W. B.

ILLINOIS

Bloomington—
MaGirl, W. J.
Nesmith, O. E.

Chicago—
Aeberly, J. J.
Arenberg, M. K.
Bailey, J. H.
Balsam, C. P.
Baumgardner, C. M.
Bennett, R. E.
Black, F. C.
Bolte, E. E.
Bracken, J. H.
Braun, L. T.
Brayton, W. M.
Broom, B. A.
Brown, A. P.
Carman, G. G.
Cartier, C. E.
Casey, B. L.
Christman, W. F.
Close, P. D.
Crone, C. E., Jr.
Currier, C. H.
Cutler, J. A.
Davis, H. H.
DeLand, C. W.
Doherty, R.
Dunham, C. A.
Emmert, L. D.
Ericsson, E. B.
Finan, J. J.
Foster, T. R.
Frank, J. M.
Gardner, W., Jr.
Gaylord, F. H.
Getschow, G. M.
Getschow, R. M.
Gibbs, F. C.
Goelz, A. H.
Gossett, E. J.
Graves, W. B.

Gray, R. F.
Grebe, H. W.
Haas, S. L.
Hagerman, J. J.
Haines, J. J.
Hale, J. F.
Hanley, T. F., Jr.
Hart, H. M.
Hartman, F. E.
Hattis, R. E.
Hayden, C. F.
Hayes, J. J.
Hayward, R. B.
Heck, G. L., Jr.
Heckel, E. P.
Herlihy, J. J.
Hill, E. V.
Hornung, J. C.
Horton, H. F.
Howatt, J.
Howell, L.
Hubbard, G. W.
Hustoel, A. M.
Jennings, W. G.
Jenson, J. S.
Johns, H. B.
Johnson, C. W.
Johnson, L. O.
Keeney, F. P.
Kehm, H. S.
Kinney, W. H.
Kirkpatrick, A. H.
Kitch, S. B.
Kreissl, H. G.
Lagodzinski, H. J.
Larson, J. M.
Lautenschlager, F.
Lees, H. K.
Lenone, J. M.
Lewis, S. R.
Love, H. G.
Maier, H. F.
Malone, D. G.
Malvin, R. C.
Marschall, P. J.
Martin, A. B.
Matchett, J. C.
Mathis, E.
Mathis, H.
Mathis, J. W.
Mathis, V. J.
Mauer, W. J.
May, M. F.
McCauley, J. H.
McClellan, J. E.
McDonnell, E. N.
McFarland, W. P.
McIlvaine, J. H.
Mertz, W. A.
Miller, F. A.
Miller, J. E.
Miller, R. T.
Milliken, J. H.
Moffitt, R. M.
Mueller, H. C.
Murphy, E. T.
Narowetz, L. L., Jr.
Needler, J. H.
Neiler, S. G.
Newport, C. F.
O'Brien, J. H.
Offen, B.
Olsen, C. F.
Pence, M. D.
Petersen, A. J.
Pitcher, L. J.
Pope, S. A.
Powers, F. W.
Prentice, O. J.
Price, C. E.
Rasmussen, R. P.
Raymond, F. I.
Reid, H. P.
Ries, L. S.
Rottmayer, S. I.
Scheidecker, D. B.
Schwelm, H. J.
Seelig, L.
Shufelt, H. M.
Shultz, E.
Smail, A. M.
Small, J. D.
Snider, L. A.
Spielman, G. P.
Stannard, J. M.
Stockenberg, R.
Sutcliffe, A. G.
Thinn, C. A.
Thomas, R. H.
Thommen, A. A.
Truitt, J. E.
Trumbo, S. M.
Vernon, J. R.
Wallace, K. S.
Walters, W. T.
Walther, V. H.
Warren, W. J.
Washington, L. W.
Weil, M.
Weil, M. I.
Weinshank, T.
Weldy, L. O.
Widdicombe, R. A.
Wilson, W. H.
Wood, F. C.
Yardley, R. W.
Zack, H. J.
Zimmerman, A. H.

Cicero—
Keppner, H. W.

Decatur—
Shorb, W. A.

Evanston—
Hayes, J. J.
Mittendorf, E. M.
Moore, R. E.

Highland Park—
Gifford, E. W.

Joliet—
Beasom, G. R.
Russell, W. B.

Homewood—
Wilkinson, F. J.

Kewanee—
Bronson, C. E.
Dickson, R. B.
Hartman, J. M.
Pursell, H. E.

LaGrange—
Linn, H. R.

Maywood—
Pelletier, A.

Moline—
Beling, E. H.
Nelson, H. W.
Nelson, R. H.
Nordheimer, L. F.
Otis, G. E.

Oak Park—
Barnes, R. B.
Blanding, G. H.
Nightingale, G. F.
Steele, A. N.
Uhlhorn, W. J.

Peoria—
Farnsworth, J. G.
Lindsey, W. J.
Meyer, F. L.

Rockford—
Braatz, C. J.
Merwin, G. E.
Stewart, D. J.

Urbana—
Broderick, E. L.
Fahnestock, M. K.
Konzo, S.
Kratz, A. P.
Severns, W. H.
Willard, A. C.

Villa Park—
Armspach, O. W.

Western Springs—
Brakenridge, C. E.

Wilmette—
Norris, W. D.

Winnetka—
Ellis, E. E.
Rietz, E. W.

Zion—
Koetz, L.

INDIANA

Evansville—
Bulleit, C. R.

Fort Wayne—
Nagelsen, L. M.

Huntington—
Smith, G. W.

Indianapolis—
Ammerman, C. R.
Fenstermaker, S. E.
Hagedon, C. H.
Hayes, J. G.
Heidenreich, G.
Kruse, R. W.
Neal, H. W.
Ott, R. C.
Poehner, R. E.
Rotz, J. M.
Shipp, C. C.
Warren, C. N.

LaPorte—
Mathis, G. A.
Shrock, J. H.

Michigan City—
Stockwell, W. R.

Peru—
Pyle, J. W.
Thrush, H. A.

St. Mary-of-the-Woods—
Bisch, B. J.

Terre Haute—
Mandel, H. J.

Wabash—
Shivers, P. F.

West Lafayette—
Hoffman, J. D.

IOWA

Ackley—
Nelson, G. O.

Cedar Rapids—
Chandler, C. W.

LeMars—
Mathey, N. J.

Sioux City—
Hagan, W. V.

Waterloo—
Winterbottom, R. F.

KANSAS

Emporia—
Burnap, C. W.

Hutchinson—
Hertz, H. P.
Mann, A. R.
Stevens, H. L.

Salina—
Ryan, W. F.

KENTUCKY

Anchorage—
Lewis, J. C.

Blandville—
Belt, N. O.

Lexington—
Anderson, F. P.
O'Bannon, L. S.
Porter, R. C.

London—
Edwards, D. C., Jr.

Louisville—
Birkholz, H. E.
Hellstrom, J.
Murphy, H. C.
Reed, W. M.
Sylvan, S. G.

LOUISIANA

New Orleans—
Gammill, O. E., Jr.
May, G. E.

Rochelle—
Abel, D. M.

MAINE

Portland—
Fels, A. B.
Merrill, C. J.
Swanson, H.

MARYLAND

Baltimore—
Axeman, J. E.
Collier, W. I.
Leilich, R. L.
McCormack, D.
McCrea, L. W.
Morris, E. J.
Posey, J.
Seiter, J. E.
Vance, L. G.
Vincent, P. J.
Whiteley, S. M.

Ellicott City—
Tibbets, J. C.

Rockville—
Brunett, A. L.

Roland Park—
Dorsey, F. C.

MASSACHUSETTS

Arlington—
Shaw, N. J. H.

Arlington Heights—
Tarr, H. M.

Belmont—
Schanze, A. G.

Boston—
Bartlett, A. C.
Berchtold, E. W.
Boyden, D. S.
Boynton, D. W.
Brinton, J. W.
Brissette, L. A.
Bryant, A. G.
Bullock, T. A.
Calvert, N. W.
Cummings, C. H.
Drinker, P.
Dusossoit, E. A.
Edwards, D. J.
Foulds, P. A. L.
Franklin, R. S.
Gleason, G. H.
Goodrich, C. F.
Hajek, W. J.
Heath, F. R.
Herrick, D. A.
Hershey, L. A.
Hilliard, C. E.
Hoyt, C. W.
Keefe, E. T.
Kelley, J. J.
Kellogg, A.
Kimball, C. W.
McCoy, T. F.
Moulton, D.
Mower, W. P.
Osborne, M. M.
Plunkett, J. H.
Rydell, C. A.
Shaw, E.
Stetson, L. R.
Stone, E. R.
Swaney, C. R.
Thompson, R. C.
Tilden, E. E.
Turner, J.
Tuttle, J. F.
Waterman, J. H.
Yaglou, C. P.

Cambridge—
Baker, R. H.
Buford, J. W.
Flint, C. T.
Haddock, I. T.
Holt, J.
Kahan, C.
Klonower, A. A.
Lees, J. T.
MacDonald, E. A.
Riley, E. C.

Cochituate—
Ahearn, W. J.

Dorchester—
Brown, M.
Hosterman, C. O.
Richard, I. T.

Fitchburg—
Karlson, A. F.

Harwich Port—
Maxwell, G. W.

Holbrook—
Nason, G. L.

Hyde Park—
Ellis, F. R.
Fritzberg, L. H.
Keyes, R. E.
Miller, J. F. G.
Wolf, J. C.

Jamaica Plain—
Jones, W. T.

Lawrence—
Bride, W. T.

Leominster—
Kern, R. T.

Lexington—
Brigham, F. H.
Burdoin, A. J.

Lynn—
Feehan, J. B.
Oates, W. A.
Reardon, J. A.

Medford—
Cushman, L. D.
Dane, I. S.

Melrose—
Cole, E. Q.
Gerrish, G. B.
Pierce, W. M.

Milton—
Mitchell, C. H.

Needham—
Park, C. D.
Webb, J. S.

Newton Centre—
Murray, J. J.

Newtonville—
Emerson, R. R.
McMurrer, L. J.

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Salisbury, W. C.

Quincy—
Gesmer, J.

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Ingalls, F. D. B.

Revere—
Foulds, S. T. N.

Roslindale—
Rehling, H. F.

Roxbury—
Lundquist, R. A.

Springfield—
Brown, W. M.
Cotter, L. F.
Cross, R. E.
Leland, W. B.
Murphy, W. W.

Watertown—
Wiegner, H. B.

Wellesley Hills—
Barnes, W. E.
Gilling, W. F., Jr.

West Roxbury—
Abboud, A.
Christie, A. Y.
McPherson, W. A.

Weymouth—
Clough, L.

Winchester—
Jackson, A. B.

Winter Hill—
Cummings, C. A.

Woburn—
Parker, P.

Wollaston—
Hodgdon, H. A.

Worcester—
Robinson, H. C.
Wechsberg, O.

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Hutzel, A. F.

Battle Creek—
Christenson, H.

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Blackmore, F. H.
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Buchanan—
Spafford, L. B.

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Barth, H. E.
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Boales, W. G.
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Brown, T.
Carroll, W. J.
Chester, T.
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Connell, R. F.
Coon, T. E.
Cooper, F. D.
Corlett, L. D.
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Davis, L. J.
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Eggleston, L. W.
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Fortune, J. R.
Gallarno, C. A.
Glanz, E.
Gordon, R. H.
Hamlin, H. A.
Hare, W. A.
Harms, W. T.
Harrigan, E. M.
Harrigan, E. R.
Harrigan, H. H.
Harrison, C. G.
Hatton, A. E.
Heydon, C. G.
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Kice, M. S., Jr.
Kilner, J. S.
Knibb, A. E.
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McColl, J. R.
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McIntire, J. F.
McLean, D.
McNair, E. E.
Milward, R. K.
Morgan, C. S.
Morse, C. T.
Olson, R. G.
Paetz, H. E.
Parrott, L. G.
Partlan, J. W.
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Purcell, F. C.
Purcell, R. E.
Randall, W. C.
Rowe, W. A.
Sauer, R. L.
Shea, M. B.
Shuell, F. W.
Snell, E.
Snyder, J. W.
Spitzley, R. L.
Spurgeon, J. H.
Toonder, C. L.
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Wallich, A. C.
Walton, H. L.
Wigle, B. M.
Winans, G. D.

Dowagiac—
Firestone, J. F.
Torr, T. W.

E. Lansing—
Miller, L. G.

Ferrysburg—
Johnston, J. F., Jr.

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Leigh, R. L.
Morton, C. H.
Taze, D. L.
Troske, J. J.
Wierenga, P. O.
Ziesse, K. L.

Hermansville—
Bernstrom, B.
Voorhees, G. A.

Highland Park—
Kintz, L. H.

Holland—
Cherven, V. W.
Sawyer, J. N.
Van Alsburg, J. H.

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Downs, S. H.
McConner, C. R.
Schlicting, W. G.
Temple, W. J.
Treadway, Q.

Lansing—
Distel, F.
Huffaker, H. B.
Parsons, R. A.

Muskegon—
Johnson, P. H.

Muskegon Heights—
Reid, H. F.

Pontiac—
Chappell, H. D.

Royal Oak—
Coughlin, R. J.
Phillips, W. J.
Turner, J. W.

St. Joseph—
Milliken, V. D.

MINNESOTA

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Duluth—
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Ten Brook, C. S.

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Bjerken, M. H.
Bradford, H. H.
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Burns, E. J.
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Cash, T. T.
Copperud, E. R.
Dahlstrom, G. A.
Dock, C. J.
Fitts, C. D.
Forfar, D. M.
Gerrish, H. E.
Gordon, E. B., Jr.
Gross, L. C.
Hall, J. R.
Hanson, L. C.
Hanson, L. P.
Harris, J. B.
Helstrom, H. G.
Hitchcock, P. C.
Huch, A. J.
Jacobson, R. A.
Johnson, L. H.
Jones, N. W.
Jordan, R. C.
King, R. L.
Kuehn, W. C.
Kuempel, L. L.
Legler, F. W.
Leslie, D. E.
Lewis, C. E.
Lowe, H. H.
Lund, C. E.
Magney, G. R.
Martenis, J. V.
McNamara, W.
Miller, L. B.
Morgan, G. C.
Noble, T. G.
Pfeifer, O. J., Jr.
Porter, H. M.
Powell, K. A.
Reese, P. R.
Rowley, F. B.
Sanford, A. L.
Stiller, F. W.
Sundell, S. S.
Swenson, J. E.
Uhl, E. J.
Uhl, W. F.
Van Horn, H. T.
Welter, M. A.

Nicollet Island—
Schernbeck, F. H.

Owatonna—
Clarkson, W. B.

Rochester—
Adams, N. D.

Shevlin—
Nesdahl, E.

St. Paul—
Arnold, E. Y.
Backstrom, R. E.
Buenger, A.
Challman, S. A.
Gausman, C. E.
Heagler, J. M.
Hickey, D. W.
Jones, E. F.
Lewis, E. B.
Lindberg, A. F.
Morton, H. S.
Oberg, H. C.
Otto, R. W.
Parks, W. N.
Ruff, D. C.
Schulte, R. R.
Swanstrom, A. E.
Winterer, F. C.
Wunderlich, M. S.

Walnut Grove—
Wiggins, O. J.

Winona—
DeLancey, R. W.

MISSOURI

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Szombathy, L. R.

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Arthur, J. M.
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Bliss, G. L.
Buckley, M. B.
Caleb, D.
Campbell, E. K.
Campbell, E. K., Jr.
Chase, L. R.
Clegg, C.
Cook, B. F.
Dawson, T. L.
Dodds, F. F.
Downes, N. W.
Fehlig, J. B.
Filkins, H. L.
Flarsheim, C. A.
Gillham, W. E.
Haas, E., Jr.
Hitchcock, F. P.
Kitchen, J. H.
Lewis, J. G.
Miller, E. S.
Millis, L. W.
Natkin, B.
Nottberg, G.
Nottberg, H.
Olchoff, M.
Olson, G. E.
Pines, S.
Rudio, H. M.
Russell, W. A.
Sheppard, F. A.
Stephenson, L. A.
Weiss, C. A.
Woodling, M. D.
Wymore, F. C.

Maplewood—
Walters, A. L.

St. Louis—
Barry, J. G., Jr.
Bayse, H. V.
Bowers, J. S.
Bradley, E. P.
Carlson, E. E.
Clegg, R. R.
Cooper, J. W.
Corrigan, J. A.
Davis, C. R.
DuBois, L. J.
Edwards, D. F.
Evleth, E. B.
Fagin, D. J.
Falvey, J. D.
Foster, J. M.
Gilmore, L. A.
Griffin, J. J.
Grossmann, H. A.
Hamilton, J. E.
Hartwein, C. E.
Hester, T. J.
Kent, J. K.
Langenberg, E. B.
McLarney, H. W.
Moon, L. W.
Myers, G. W. F.
Reed, P. L.
Rodenheiser, G. B.
Rosebrough, R. M.
Sachleben, E. H.
Seepe, P. E.
Sodemann, P. W.
Sodemann, W. C. B.
Stammer, E. L.
Tenkonohy, R. J.
White, E. A.
Wolff, O. H.

Webster Groves—
Axthelm, F. G.
Gabelman, H. D.
Ranck, G. L.

MONTANA

Big Timber—
Strickland, A. W.

Billings—
Cohagen, C. C.
Tooker, C. C.

Bozeman—
Powers, F. I.

NEBRASKA

Clarks—
Manning, W. M.

Omaha—
Eaton, B. K.
Lewis, W.
Sebree, G. M.

Scotts Bluff—
Davis, O. E.

NEW HAMPSHIRE

Lebanon—
Lewis, J. P.

NEW JERSEY

Arlington—
Adler, A. A.
Bock, B. A.
Schwertfeger, A.

Atlantic City—
Labov, M.
Strouse, S. B.

Bayonne—
Schwartz, J.

Belmar—
Merkel, F. P.

Bloomfield—
Hochuli, H. W.

Camden—
Brown, W. M.
Kappel, G. W. A.
Lanning, E. K.
Webster, E. K.
Webster, W.
Webster, W., Jr.

East Orange—
Gombers, H. B.
Grahn, V. F.
Rack, E. C.
Reilly, J. H.
Schroth, A. H.
Tallmadge, W.
Turno, W. G. W.
Worsham, H.

Elizabeth—
Burke, J. J.
Cornwall, G. I.
Grant, W. A.
Lyman, S. E.
Skidmore, J. G.
Taylor, K. A.
Wheller, H. S.

Essex Fells—
Carrier, E. G.
Stacey, A. E., Jr.

Fair Haven—
Smith, H. P.

Grantwood—
Butler, P. D.

Haddonfield—
Dobbs, C. E.
Jones, R. E.

Hackensack—
Turnau, E. H.

Hasbrouck Heights—
Goodwin, S. L.

Hawthorne—
Lawton, F. C.

Hillside—
Tichenor, L. R., Jr.

Irvington—
Freas, R. B.
Reinke, A. G.

Jersey City—
Dunbrack, A. P.
Hashagen, J. B.
Jones, H. L.
Kelly, C. J.
Ritchie, W.
Van Court, W. G.
Walterthum, J. J.

Jobstown—
Allinson, O. H.

Keansburg—
Roske, F. M.

Long Branch—
Hammond, M. J.

Maplewood—
Ehrlich, M. W.
Evans, W. A.
Smith, M. S.

Merchantville—
Binder, C. G.
Rohlin, K. W.

Montclair—
Bentz, H.
Holbrook, F. M.
Long, D. R.
Osmundsen, H. B.

Newark—
Alt, H. L.
Ashley, C. M.
Atkinson, K. B.
Bryant, P. J.
Bynum, O. W.
Carey, P. C.
Carrier, W. H.
Day, V. S.
Finney, G. J.
French, D. E.
Harris, C. R.
Hirst, J. N.
Holton, J. H.
Ingels, M.
Leinroth, J. P.
Lewis, J. L.
Lewis, T.
Lyle, J. I.
Matullo, J. R.
Morehouse, H. P.
Rachal, J. M.
Ray, L. B.
Soule, L. C.
Tenney, D.
Thornburg, H. A.
West, P.

North Arlington—
Bermel, A. H.
Emery, H.
Vogelbach, O.

Palmyra—
Cassell, J. D.

Paterson—
Cox, H. F.
Pryor, F. L.

Perth Amboy—
Simkin, M.

Plainfield—
Hedges, H. B.
Tobin, G. J.

Ridgefield Park—
Davis, A. C.

Ridgewood—
Fitts, J. C.

Riverton—
Brunt, T. B.

Rutherford—
Nelson, C. L.

Short Hills—
Fouilhoux, J. A.

South Orange—
Hansen, C. C.

Summit—
Oaks, O. O.

Teaneck—
Heebner, W. M.

Tenafly—
Redfield, C.

Union City—
Taverna, F. F.

Upper Montclair—
Armagnac, A. S.
Fernald, H. B., Jr.

Verona—
Stone, G. F.

West Collingswood—
Knecht, C. H.

NEW YORK

Albany—
Anker, G. W.
Bond, H. A.
Johnson, H. S.
Murray, T. F.
Ryan, H. J.
Taggart, R. C.
Teeling, G. A.

Binghamton—
Brown, R. F.
Marum, O.

Bronxville—
Dornheim, G. A.

Buffalo—
Barnett, S. J.
Beman, M. C.
Booth, C. A.
Burke, F. H.
Candee, B. C.
Cherry, L. A.
Cheyney, C. C.
Corbin, W. E.
Cressy, R. E.
Criqui, A. A.
Danforth, N. L.
Dyer, O. K.
Erdle, G. F.
Evans, C. A.
Farnham, R.
Farrar, C. W.
Fraser, W. G.
Harding, L. A.
Heath, W. R.
Hedley, P. S.
Hexamer, H. D.
Jackson, M. S.
Johnson, E. E.
Kamman, A. R.
Landers, J. J.
Love, C. H.
Madison, R. D.
Mahoney, D. J.
McTernan, F. J.
Mosher, C. H.
Rente, H. W.
Roebuck, W., Jr.
Seelbach, H.
Shelney, T.
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Thornton, R. T.
Voisinet, W. E.
Wendt, E. F.
Yager, J. J.

Eggertsville—
Eiss, R. M.
Hirschman, W. F.

Elmira—
Davis, B. C.
McGlenn, G. R.

Geneva—
Herendeen, F. W.

Hamburg—
Dempsey, H. P.

Hastings-on-Hudson—
Reynolds, T. W.

Hornell—
Brill, J. W.

Hudson Falls—
Hollister, E. W.

Irvington-on-Hudson—
Bastedo, A. E.
Knox, J. R.

Ithaca—
Barns, A. A.
Sawdon, W. M.
Williams, J. W.

Jamestown—
Sharp, F. H.

Kenmore—
Davis, J.
Green, J. J.
Krueger, B.
Quigley, W. J.
Rente, S. R.
Roseberry, J. H.
Snyder, J. S.

Larchmont—
Beirn, J. U.
Gaylor, W. S.

LeRoy—
Rinehart, W. R.

Lockport—
Bishop, C. R.
Harrison, A. B. B.
Saunders, L. P.

Morton—
Stangland, B. F.

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Northon, L.
Obert, C. W.
Ryan, J. E.

New Rochelle—
Abrams, A.
Farley, W. F.

New York City—
Addams, H.
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Alvord, A. M.
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Atherton, G. R.
Austin, H. F., Jr. (Patchogue, L. I.)
Bachler, L. J.
Bampton, C. M. (Brooklyn)
Barbera, H. A. (Corona, L. I.)
Barbieri, P. J.
Barnum, M. C.
Baum, A. L.
Beebe, F. E. W.
Bennett, E. A.
Bennitt, G. E.
Berman, L. K.
Bernhard, G. (Lawrence, L. I.)
Birch, H. R.
Birukoff, R. R.
Blackburn, E. C., Jr. (Hempstead, L. I.)
Blackman, A. O.
Blackmore, J. J.
Blitz, E.
Bodinger, J. H.
Bolton, R. P.
Bowles, P.
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Brown, W. A.
Browne, A. L.
Bruckmann, J. C.
Buensod, A. C.
Bulkeley, C. A.
Burbaum, W. A.
Callaghan, P. F., Jr. (Brooklyn)
Campbell, F. B.
Carpenter, R. H.
Chase, C. L. (Brooklyn)
Cohagen, N.
Cooper, J. R.
Crone, T. E.

Cucci, V. J.
Dailey, J. A.
(Astoria, L. I.)
Daly, R. E.
Darts, J. A.
Donnelly, R.
Downe, E. R.
Dresen, W. D.
(Jackson Heights, L. I.)
Driscoll, W. H.
Duff, K.
Duffield, T. J.
Durkee, M. E.
Duryea, A. A.
(City Island)
Dwyer, T. F.
(Brooklyn)
Eadie, J. G.
Easterbrooks, C. C.
Eells, H. B.
(Brooklyn)
Elliott, L.
Engle, A.
Everetts, J., Jr.
Fansler, P. E.
Fay, F. C.
Fegley, D. R.
Feldman, A. M.
Fenner, N. P.
Fenton, F. A.
Ferrero, H. J.
Fife, G. D.
Finch, S. B.
(Brooklyn)
Fleisher, W. L.
Flink, C. H.
Frank, O. E.
Friedman, A.
Friedman, M.
Gautesen, A.
Genchi, B.
(Brooklyn)
Giannini, A. A.
Gilmour, A. B.
(Brooklyn)
Glore, E. F.
Goldschmidt, O. E.
Gornston, M. H.
(Brooklyn)
Goulding, W.
(Brooklyn)
Griffin, B. H.
(Stony Brook L.I.)
Grossman, H. E.
(Oceanside, L. I.)
Haigney, J. E.
Hamburger, F. G.
Hament, L.
Hamburger, F. W.
Hartman, F. S.
Heibel, W. E.
Henry, A. S., Jr.
Herty, F. B.
(Brooklyn)
Hertzler, J. R.
(Brooklyn)
Higgins, T. J.
Hinchman, E. G.
(Brooklyn)
Hinkle, E. C.
(Hempstead, L. I.)
Hinrichsen, A. F.
Hoffman, C. S.
Hollister, N. A.
(Brooklyn)
Hosking, H. L.
Hotchkiss, C. H. B.
Howell, F. B.
Hyman, W. M.
Issertell, H. G.
Jacobus, D. S.
Jalonack, I. G.
(Patchogue, L. I.)
Janco, N.
Janet, H. L.

Johnson, E. B.
(W. New Brighton, S. I.)
Johnston, W. H.
Kagey, I. B., Jr.
Kastner, G. C.
Kaufman, W. M.
Kenward, S. B.
(Bayshore)
Keplinger, W. L.
Kiewitz, A. A.
(Astoria, L. I.)
Kimball, D. D.
Kingsley, E. A.
Kirk, L. G.
Knopf, C.
Koithan, W. S.
Kreitner, W. G.
(Brooklyn)
Kroupsky, V.
Kuhlmann, R.
Le Beau, J. F.
(Jamaica, L. I.)
Lefingwell, R. R.
Lennon, J. O.
Leupold, H. W.
(Brooklyn)
Lowy, M. R.
Lucke, C. E.
Lyle, E. T.
Lyon, P. S.
Maiman, H.
(Glendale, L. I.)
Mandeville, E. W.
(Brooklyn)
Marino, D. A.
Markush, E. U.
Marshall, H. H.
Martin, G. W.
McKelvey, D. M.
McKiever, W. H.
McLeish, W. S.
Mehne, C. A.
Meinke, H. G.
Meisel, C. L.
Meyer, C. L.
(Hollis, L. I.)
Meyer, H. C., Jr.
Miller, C. A.
Monroe, M.
Montgomery, O. C.
Moore, R. E.
(Brooklyn)
Moss, E.
(Brooklyn)
Munder, J. F. ,Jr.
Munier, L. L.
Munro, E. A.
(Lynbrook L. I.)
Neale, L. I.
Neary, D. A.
Neideck, A. A.
Nelson, G. A.
Nicol, N. C.
Offner, A. J.
O'Hare, G. W., Jr.
Olsen, G. E.
(Arverne, L. I.)
Olstad, M. H.
Olvany, W. J.
Osburn, R. M.
Patorno, S. A. S.
Paulding, L. G.
Perina, A. E.
(Staten Island)
Pfuhler, J. L.
(W. New Brighton, (S. I.)
Phillips, F. W., Jr.
Pihlman, A. A.
Pinder, P. H.
Pison, D., Jr.
Place, C. R.
Pohle, K. F.
Presdee, C. W.

Price, E. H.
(Riverhead, L. I.)
Purdy, R. B.
Purinton, D. J.
Quirk, C. H.
Raffes, A.
Raisler, R. K.
Reed, J. F.
Reynolds, W. V.
Richardson, H. T.
Riley, C. L.
Ritchie, E. J.
Ritter, A.
Rodman, R. W.
Rosenberg, P.
Ross, J. O.
Roth, C. F.
Rothrock, J. T.
(Douglaston, L.I.)
Rozett, W., Jr.
Ruppert, E. H.
(Brooklyn)
Schleyer, E. F.
(Brooklyn)
Schlossman, M. B.
Schneider, W. G.
Schoepflin, P. H.
Schulze, B. H.
Scott, C. E.
Scott, G. M.
Scribner, E. D.
Seelig, A. E.
Sellman, N. T.
Senior, R. L.
Seward, P. H.
(Brooklyn)
Shepard, E. C.
Siebs, C. T.
Siegel, L.
(Brooklyn)
Sklenarick, L.
Smith, P. C., Jr.
(Northport, L. I.)
Stack, F. C.
(Flushing, L. I.)
Staples, W. H.
Stern, H. R.
Sternberg, E.
Stewart, C. W.
Still, F. R.
Strock, C.
Strunin, J.
Sullivan, D. A.
Sutton, F.
Syska, A. G.
Taylor, J. H.
(Queens Village, (L. I.)
Thomson, T. N.
(Huntington, L.I.)
Tiltz, B. E.
Timmis, W. W.
Tisnower, W.
Torrance, H.
Tucker, F. N.
(Freeport, L. I.)
Tusch, W.
(Brooklyn)
Tyler, R. D.
Vetlesen, G. U.
Vivarttas, E. A.
(Brooklyn)
Vogt, J. H.
Wachs, L. J.
(Brooklyn)
Waechter, H. P.
(Tompkinsville, S. I.)
Wallace, G. J.
(East Elmhurst, (L. I.)
Wallace, W. M., II
(Hollis, L. I.)
Waring, J. M. S.
Wells, E. E.
(Beechhurst, L. I.)

White, E. S.
Whitelaw, H. L.
Wilder, E. L.
Willis, R. C.
Wilson, H. A., Jr.
Winquist, W. J.
(Brooklyn)
Wolff, R. A.
Yockel, T. J.

North Tarrytown—
Deely, J. J.

Ogdensburg—
Skelly, J. F.

Oriskany—
Oakey, W. E.

Ossining—
Hooper, V. F.

Pelham Manor—
Peacock, J. K.

Port Jervis—
Trimmer, C. M.

Poughkeepsie—
Doherty, J. J.

Rochester—
Cherne, R. E.
Coe, R. T.
Cook, R. P.
Hakes, L. M.
Sheldon, N. E.
Stacy, S. C.
Welsh, H. S.

Rome—
Lynch, W. L.
Steele, M. G.

Scarsdale—
Cumming, R. W.
Fiedler, H. W.
Ullman, H. G.

Schenectady—
Cannon, C. N.
Faust, F. H.
Harrington, E. D.
McLenegan, D. W.
Stevenson, A. R., Jr.
Strachan, J. S.
Vogel, A.
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Silver Creek—
Giguere, G. H.

Snyder—
John, V. P.

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Acheson, A. R.
Cook, C. J.
Ormsby, H. K., Jr.

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Abraham, L.
Weiss, A. P.

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Vogt, J. B.

Utica—
Steinhorst, T. F.

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Zibold, C. E.

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Kelly, J. G.
Rainger, W. F.
Stitt, A. B.
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Small, B. R.

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Bahnson, F. F.

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Strobl, J. A.

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Myers, F. L.

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Black, E. N., 3rd

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Sherry, R. W.

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Slight, I.

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Dockeray, F.
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Schrader, C. C.
Yeomans, P. H.

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James, H. R.

McKeesport—
Dugan, T. M.

McKees Rocks—
Bucher, H. G.

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Williams, L. E.

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Atkins, T. J.

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Wilmot, C. S.

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Kaczenski, C.

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Aronson, H. H.
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Black, H. G.
Blankin, M. F.
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Boon, G.
Bornemann, W. A.
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Caldwell, A. C.
Clarkson, R. C., Jr.
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Cornell, J. C.

Culbert, W. P.
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Davidson, L. C.
Davidson, P. L.
Donovan, W. J.
Eagan, W. H.
Eakins, W.
Eggly, H. J., Jr.
Eichberg, W. R.
Elliot, E.
Engel, E.
Erickson, H. H.
Ewald, W.
Faltenbacher, H. J.
Familetti, A. R.
Fest, L. T.
Galligan, A. B.
Gant, H. P.
Gillett, M. C.
Glassey, J. W.
Hackett, H. B.
Harris, J. E.
Hibbs, F. C.
Hoft, P. J.
Hunger, R. F.
Hurley, J. C.
Hynes, L. P.
Ickeringill, J.
Jellett, S. A.
Johnson, C. E.
Keeler, A. H.
Kelble, F. R.
Kelly, J. A.
Kerney, T. F.
Kriebel, A. E.
Kuehnert, A. C.
Lewis, G. C.
Liner, J. J.
Lutz, P. R.
MacDade, A. H.
Mann, L. B.
Marks, A. A.
Mather, H. H.
McCarthy, C. J.
McClintock, A., Jr.
Mellon, J. T. J.
Mensing, F. D.
Meyer, J. W.
Miller, A. A.
Monday, C. E.
Morgan R. C.
Naylor, C. L.
Nesbitt, A. J.
Nesbitt, J. J.
Nusbaum, L.
Plass, C. W.
Plewes, S. E.
Pryibil, P. L.
Redstone, A. L.
Reilly, C. E.
Rettew, H. F.
Reuss, E. H., Jr.
Rhea, C. A.
Roberts, E. F., Jr.
Rugart, K.
Sabin, E. R.
Sanbern, E. N.
Sands, J. S.
Shanklin, A. P.
Sheffler, M.
Shepard, J. deB.
Speckman, C. H.
Stevens, J. M.
Stoever, G. H.
Timmis, P.
Touton, R. D.
Traugott, M.
Tuckerman, G. E.
Wandless, F. W.
Wegmann, A.
Welamb, V. N.
White, J. J.
Why, H. B.
Wild, W. H.
Wilson, B. W.
Woolston, A. H.

Pittsburgh—
Alcott, W. L.
Aston, J.
Babbitt, W. D.
Beighel, H. A.
Benson, M. A.
Blackmore, G. C.
Blackmore, J. S.
Blackshaw, J. L.
Bowers, R. C.
Brauer, R.
Brewer, J. G.
Burns, J. R.
Bushnell, C. D.
Carr, M. L.
Collins, J. F. S., Jr.
Dice, E. S.
Digby, H. E.
Dorfan, M. I.
Edwards, P. A.
Evans, E. C.
Fitzsimons, J. P.
Fritz, C. V.
Gunther, F. A.
Hanson, E. W.
Harper, S. H.
Hecht, F. H.
Heilman, R. H.
Hickey, J. W.
Houghten, F. C.
Humphreys, C. M.
Hunter, V. W.
Ingold, J. W.
Keist, W. E.
Kellner, D. C.
Kennedy, O. A.
Lutz, W. J.
Maehling, L. S.
Maginn, P. F.
McClanahan, L. C.
McGinness, J. E.
McGonagle, A.
McGuigan, L. A.
McIntosh, F. C.
Miller, R. A.
Nass, A. F.
Nicholls, P.
Nordheimer, C. L.
O'Neill, P.
Orr, H. B.
Pennel, R.
Pittock, L. B.
Reed, V. A., Jr.
Richmond, J.
Riddle, K. L.
Riesmeyer, E. H., Jr.
Rockwell, J. F.
Smith, R. H.
Smyers, E. C.
Speller, F. N.
Stanger, R. B.
Steen, J. M.
Stevenson, W. W.
Tennant, R. J. J.
Tower, E. S.
Vinson, N. L.
Waters, G. G.
Wright, C. E.

Pottsville—
Marty, E. O.
Smith, J. D.

Reading—
Luck, A. W.
Nicely, J. E.

Rutledge—
Vroome, A. E.

Scranton—
Gilboy, J. P.
Shaver, H. H.

Sewickley—
Black, G. E.

State College—
Queer, E. R.

Stroudsburg—
Kiefer, E. J., Jr.

Tamaqua—
Hadesty, A. L., Jr.
Koch, H. O.

Upper Darby—
Eastman, C. B.
Taliaferro, R. R.

Vernon—
Etheridge, G. T., Jr.

Villanova—
Barr, G. W.
Carey, J. A.

Wayne—
Patrick, H. M.

Wilkes-Barre—
Santee, H. C.

Wilkinsburg—
Campbell, T. F.
Vernier, M. G.

Williamsport—
Pfeiffer, J. F.

Wormleysburg—
Miller, T. G.

York—
Baker, I. C.

RHODE ISLAND

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Wyman, D. M.

Providence—
Coleman, J. B.
Gibbs, E. W.
Hartwell, J. C.
Husband, E. W.
McLaughlin, J. D.
Moulder, A. W.

Washington—
Wilson, H. A.

TENNESSEE

Memphis—
Campbell, A. Q., Jr.
D'Imor, E. J.
Hoshall, R. H.

Nashville—
Brown, F.
Jarratt, P. R.

TEXAS

Amarillo—
Burnett, E. S.

Austin—
Groseclose, J. B.

College Station—
Badgett, W. H.
Giesecke, F. E.
Reynolds, J. A.
Smith, E. G.

Dallas—
Cunningham, T. M.
Durning, E. H.
Hersh, F. C.
Landauer, L. L.
Lynn, J. H.
Meffert, G. H.
Renouf, E. P.

Fort Worth—
Skinner, H. W.

Houston—
Kiesling, J. A.

Irving—
Moler, W. H.

Kingsville—
Richtmann, W. M.

Port Arthur—
Hansen, C. J.

San Antonio—
Diver, M. L.
Ebert, W. A.

VERMONT

Burlington—
Austin, F. L.
Lanou, J. E.
Raine, J. J.

North Ferrisburg—
Breckenridge, L. P.

VIRGINIA

Clarendon—
MacMahon, W. K.

Danville—
Farley, W. S.

East Falls Church—
Casey, H. F.

Lynchburg—
Doering, F. L.
Wiley, E. C.

Newport News—
Noland, L. U.

Norfolk—
Nowitzky, H. S.
Peebles, J. K., Jr.

Richmond—
Carle, W. E.
Johnston, J. A.
Livingston, B. B.
Schulz, H. I.

WASHINGTON

Seattle—
Beggs, W. E.
Bouillon, L.
Carsten, W. H.
Cook, H. A.
Cox, W. W.
Davis, R. J.
Dudley, W. L.
Eastwood, E. O.
Granston, R. O.
Griffin, D. C.
Hauan, M. J.
MacLeod, K. F.
Mallis, W.
O'Connell, P. M.
Peterson, S. D.
Pollard, A. L.
Twist, C. F.
Whaley, R. S.
Zokelt, C. G.

Tacoma—
May, C. W.
Spofforth, W.

Yakima—
McCune, B. V.

WEST VIRGINIA

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Anderson, S. W., Jr.

Charleston—
Shanklin, J. A.

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Donnelly, J. A.
Innis, H. R.

Morgantown—
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WISCONSIN

Ashland—
Trulson, A. F.

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Shodron, J. G.

Kohler—
Hvoslef, F. W.
Kohler, W. J., Jr.

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Anderegg, R. H.
Miller, M. W.
Trane, R. N.

Madison—
Dean, C. L.
Hess, D. K.
James, J. W.
Larson, G. L.
Nelson, D. W.
Plaenert, A. B.
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Menomonie—
Schoenoff, A. E.

Mondovi—
Lofte, J. A.

Milwaukee—
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Bowers, A. F.
Brown, W. H.
Ellis, H. W.
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Hanley, E. V.
Haupt, H. F.
Herrmann, H. C.
Ine, F. H.
Jackson, C. H.
Jones, E. A.
Jung, J. S.
Juttner, O. J.
Miller, C. W.
Miller, H. M.
Noll, W. F.
Peters, H. H.
Randolph, C. H.
Rice, C. J.
Shawlin, W. C.
Spielmann, H. J.
Szekely, E.
Volk, J. H.
Wagner, A. M.
Weimer, F. G.
Wilson, W. H.

Racine—
Dixon, A. G.
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Waite, H.

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Page, H. W.

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Erickson, M. E.

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Brussels—
Mautsch, R.

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Brockville, Ont.—
Davenport, R. F.

Calgary, Alberta—
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Walker, A.

Edmonton, Alberta—
Kelly, H.

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Eagar, R. F.

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Best, M. W.
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Islington, Ont.—
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Beavers, G. R.

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Montreal, West, P. Q.—
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St. Catharines, Ont.—
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CHINA

Dairen, Manchuria—
Katsumoto, E.

Nanking—
Loo, P. Y.

Osaka—
Fukui, K.

Shanghai—
Doughty, C. J.
Hart-Baker, H. W.
Kwan, I. K.
Loh, N. S.
Merritt, C. J.
Morrison, C. B.
Waung, T. F.

CZECHOSLOVAKIA

Prague—
Brust, O.

DENMARK

Copehagen—
Reck, W. E.

ENGLAND

Buckinghamshire—
Rose, W. K.

Leeds—
Jennins, H. H.

London—
Bailey, W. M.
Butt, R. E. W.
Haden, G. N.
Herring, E.
Nobbs, W. W.
Overton, S. H.

Middlesex—
Case, W. G.

Stockport—
Webb, J. W.

Swinton—
Yates, W.

Trowbridge—
Haden, W. N.

Westminster—
Russell, J. N.

Wolverhampton—
Tyson, W. H.

FRANCE

Dijon—
Bur, J. R. C.

Lille—
Neu, H. J. E.

Lyon—
Goenaga, R. C.

Paris—
Beaurrienne, A.
Downe, H. S.
Modiano, R.
Nessi, A.
Schmutz, J.

GERMANY

Berlin—
Brandi, O. H.

Stuttgart—
Jens, J.
Klein, A.

INDIA

Calcutta—
Casperd, H. W. H.

New Delhi—
Heard, J. A. E.

IRELAND

Cork—
Barry, P. I.

ITALY

Milan—
Donzelli, E.
Gini, A.

JAPAN

Tokyo—
Kitaura, S.
Kozu, T.
Saito, S.
Sekido, K.

MEXICO

Lomas de Chapultepec—
Gilfrin, G. F.
Martinez, J. J.

Mexico City—
Croft, T.

NEW ZEALAND

Auckland—
Wallace, B.

Christchurch—
Vale, H. A. L.

Dunedin—
Davies, G. W.

NORWAY

Oslo—
Alfsen, N.
Tjersland, A.

SOUTH AMERICA

Santiago, Chile—
Carrasco, S.

SWEDEN

Stockholm—
Gille, H. B.
Theorell, H. G. T.

U. S. S. R.

Leningrad—
Sakouta, M. L.
Teterevnikoff, N. N.

PAST OFFICERS

AMERICAN SOCIETY *of* HEATING *and* VENTILATING ENGINEERS

1894

President..........Edward P. Bates
1st Vice-President..........Wm. M. Mackay
2nd Vice-President..........Wiltsie F. Wolfe
3rd Vice-President..........Chas. S. Onderdonk
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, Fred P. Smith
Henry Adams — A. A. Cary
Hugh J. Barron — James A. Harding
Edward P. Bates, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter
Albert A. Cryer — Chas. W. Newton
F. W. Foster — Ulysses G. Scollay, *Secy.*

1895

President..........Stewart A. Jellett
1st Vice-President..........Wm. M. Mackay
2nd Vice-President..........Chas. S. Onderdonk
3rd Vice-President..........D. M. Quay
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, James A. Harding
Geo. B. Cobb — Ulysses G. Scollay
Wm. McMannis — B. F. Stangland
Stewart A. Jellett, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, R. C. Carpenter
Henry Adams — T. J. Waters
Edward P. Bates — Albert A. Cryer, *Secy.*

1896

President..........R. C. Carpenter
1st Vice-President..........D. M. Quay
2nd Vice-President..........Edward P. Bates
3rd Vice-President..........F. W. Foster
Treasurer..........Judson A. Goodrich
Secretary..........L. H. Hart

Board of Managers

Chairman, Wm. M. Mackay
Hugh J. Barron — Stewart A. Jellett
W. S. Hadaway, Jr. — Wiltsie F. Wolfe
R. C. Carpenter, *Pres.* — L. H. Hart, *Secy.*

Council

Chairman, A. A. Cary
Albert A. Cryer — B. F. Stangland
Wm. McMannis — J. J. Blackmore, *Secy.*

1897

President..........Wm. M. Mackay
1st Vice-President..........H. D. Crane
2nd Vice-President..........Henry Adams
3rd Vice-President..........A. E. Kenrick
Treasurer..........Judson A. Goodrich
Secretary..........H. M. Swetland

Board of Managers

Chairman, R. C. Carpenter
Edward P. Bates — Stewart A. Jellett
W. S. Hadaway, Jr. — Wiltsie F. Wolfe
Wm. M. Mackay, *Pres.* — H. M. Swetland, *Secy.*

Council

Chairman, Albert A. Cryer
John A. Fish — James Mackay
Wm. McMannis — B. F. Stangland

1898

President..........Wiltsie F. Wolfe
1st Vice-President..........J. H. Kinealy
2nd Vice-President..........A. E. Kenrick
3rd Vice-President..........John A. Fish
Treasurer..........Judson A. Goodrich
Secretary..........Stewart A. Jellett

Board of Managers

Chairman, Wm. M. Mackay
Thomas Barwice — A. C. Mott
John A. Connolly — Francis A. Williams
Wiltsie F. Wolfe, *Pres.* — Stewart A. Jellett, *Secy.*

Council

Chairman, R. C. Carpenter
Henry Adams — W. S. Hadaway, Jr.
Albert A. Cryer — Wm. McMannis
Wiltsie F. Wolfe, *Pres.* — Stewart A. Jellett, *Secy.*

1899

President..........Henry Adams
1st Vice-President..........D. M. Quay
2nd Vice-President..........A. E. Kenrick
3rd Vice-President..........Francis A. Williams
Treasurer..........Judson A. Goodrich
Secretary..........Wm. M. Mackay

Board of Managers

Chairman, Stewart A. Jellett
B. H. Carpenter — Wm. Kent
A. A. Cary — Wiltsie F. Wolfe
Henry Adams, *Pres.* — Wm. M. Mackay, *Secy.*

Council

Chairman, R. C. Carpenter
John Gormly — Wm. McMannis
W. S. Hadaway, Jr. — B. F. Stangland
Henry Adams, *Pres.* — Wm. M. Mackay, *Secy.*

1900

President....................D. M. Quay
1st Vice-President....................A. E. Kenrick
2nd Vice-President....................Francis A. Williams
Treasurer....................Judson A. Goodrich
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, D. M. Quay
Wm. Kent, *Vice-Chm.* — D. M. Nesbit
R. C. Carpenter — C. B. J. Snyder
John Gormly — Wm. M. Mackay, *Secy.*

1901

President....................J. H. Kinealy
1st Vice-President....................A. E. Kenrick
2nd Vice-President....................Andrew Harvey
Treasurer....................Judson A. Goodrich
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, J. H. Kinealy
Wm. Kent, *Vice-Chm.* — John Gormly
R. C. Carpenter — C. B. J. Snyder
R. P. Bolton — Wm. M. Mackay, *Secy.*

1902

President....................A. E. Kenrick
1st Vice-President....................Andrew Harvey
2nd Vice-President....................Robert C. Clarkson
Treasurer....................Judson A. Goodrich
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, A. E. Kenrick
John Gormly, *Vice-Chm.* — J. H. Kinealy
R. C. Carpenter — C. B. J. Snyder
Wm. Kent — Wm. M. Mackay, *Secy.*

1903

President....................H. D. Crane
1st Vice-President....................Wm. Kent
2nd Vice-President....................R. P. Bolton
Treasurer....................Judson A. Goodrich
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, H. D. Crane
C. B. J. Snyder, *Vice-Chm.* — A. E. Kenrick
R. C. Carpenter — Geo. Mehring
John Gormly — Wm. M. Mackay, *Secy.*

1904

President....................Andrew Harvey
1st Vice-President....................John Gormly
2nd Vice-President....................Robert C. Clarkson
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, Andrew Harvey
John Gormly — H. D. Crane
Robert C. Clarkson — A. E. Kenrick
J. J. Blackmore — C. B. J. Snyder
R. C. Carpenter — Wm. M. Mackay, *Secy.*

1905

President....................Wm. Kent
1st Vice-President....................R. P. Bolton
2nd Vice-President....................C. B. J. Snyder
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, Wm. Kent
R. P. Bolton — James Mackay
C. B. J. Snyder — B. F. Stangland
B. H. Carpenter — J. C. F. Trachsel
A. B. Franklin — Wm. M. Mackay, *Secy.*

1906

President....................John Gormly
1st Vice-President....................C. B. J. Snyder
2nd Vice-President....................T. J. Waters
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, John Gormly
C. B. J. Snyder, *Vice-Chm.* — James Mackay
R. C. Carpenter — B. F. Stangland
Frank K. Chew — T. J. Waters
A. B. Franklin — Wm. M. Mackay, *Secy.*

1907

President....................C. B. J. Snyder
1st Vice-President....................James Mackay
2nd Vice-President....................Wm. G. Snow
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, C. B. J. Snyder
James Mackay, *Vice-Chm.* — Frank K. Chew
R. E. Atkinson — A. B. Franklin
R. C. Carpenter — Wm. G. Snow
Edmund F. Capron — Wm. M. Mackay, *Secy.*

1908

President....................James Mackay
1st Vice-President....................Jas. D. Hoffman
2nd Vice-President....................B. F. Stangland
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, James Mackay
Jas. D. Hoffman, *Vice-Chm.* — John F. Hale
B. F. Stangland — August Kehm
R. C. Carpenter — C. B. J. Snyder
Frank K. Chew — Wm. M. Mackay, *Secy.*

1909

President....................Wm. G. Snow
1st Vice-President....................August Kehm
2nd Vice-President....................B. S. Harrison
Treasurer....................Ulysses G. Scollay
Secretary....................Wm. M. Mackay

Board of Governors

Chairman, Wm. G. Snow
August Kehm, *Vice-Chm.* — Samuel R. Lewis
John R. Allen — James Mackay
R. C. Carpenter — B. F. Stangland
B. S. Harrison — Wm. M. Mackay, *Secy.*

1910

President....Jas. D. Hoffman
1st Vice-President....R. P. Bolton
2nd Vice-President....Samuel R. Lewis
Treasurer....Ulysses G. Scollay
Secretary....Wm. M. Mackay

Board of Governors

Chairman, Jas. D. Hoffman

R. P. Bolton, *Vice-Chm.*
Geo. W. Barr
R. C. Carpenter
Judson A. Goodrich
John F. Hale
Samuel R. Lewis
James Mackay
Wm. M. Mackay, *Secy.*

1911

President....R. P. Bolton
1st Vice-President....John R. Allen
2nd Vice-President....A. B. Franklin
Treasurer....Ulysses G. Scollay
Secretary....Wm. W. Macon

Board of Governors

Chairman, R. P. Bolton

John R. Allen, *Vice-Chm.*
John T. Bradley
R. C. Carpenter
James H. Davis
A. B. Franklin
Jas. D. Hoffman
August Kehm
Wm. W. Macon, *Secy.*

1912

President....John R. Allen
1st Vice-President....John F. Hale
2nd Vice-President....Edmund F. Capron
Treasurer....James A. Donnelly
Secretary....Wm. W. Macon

Board of Governors

Chairman, John R. Allen

John F. Hale, *Vice-Chm.*
Edmund F. Capron
R. P. Bolton
Jas. D. Hoffman
Dwight D. Kimball
Samuel R. Lewis
Wm. M. Mackay
Wm. W. Macon, *Secy.*

1913

President....John F. Hale
1st Vice-President....A. B. Franklin
2nd Vice-President....Edmund F. Capron
Treasurer....James A. Donnelly
Secretary....Edwin A. Scott

Board of Governors

Chairman, John F. Hale

A. B. Franklin, *Vice-Chm.*
John R. Allen
Edmund F. Capron
R. P. Bolton
Frank T. Chapman
Ralph Collamore
James A. Donnelly
Dwight D. Kimball
Wm. W. Macon
James M. Stannard
Theodore Weinshank
Edwin A. Scott, *Secy.*

1914

President....Samuel R. Lewis
1st Vice-President....Edmund F. Capron
2nd Vice-President....Dwight D. Kimball
Treasurer....James A. Donnelly
Secretary....J. J. Blackmore

Council

Chairman, Samuel R. Lewis

E. F. Capron, *Vice-Chm.*
Dwight D. Kimball
John R. Allen
Frank T. Chapman
Frank I. Cooper
James A. Donnelly
John F. Hale
Harry M. Hart
Frank G. McCann
Wm. W. Macon
James M. Stannard
J. J. Blackmore, *Secy.*

1915

President....Dwight D. Kimball
1st Vice-President....Harry M. Hart
2nd Vice-President....Frank T. Chapman
Treasurer....Homer Addams
Secretary....J. J. Blackmore

Council

Chairman, Dwight D. Kimball

Harry M. Hart, *Vice-Chm.*
Homer Addams
Frank T. Chapman
Frank I. Cooper
E. Vernon Hill
Wm. M. Kingsbury
Samuel R. Lewis
Frank G. McCann
J. T. J. Mellon
Henry C. Meyer, Jr.
Arthur K. Ohmes
J. J. Blackmore, *Secy.*

1916

President....Harry M. Hart
1st Vice-President....Frank T. Chapman
2nd Vice-President....Arthur K. Ohmes
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Harry M. Hart

F. T. Chapman, *Vice-Chm.*
Homer Addams
Charles R. Bishop
Frank I. Cooper
Milton W. Franklin
E. Vernon Hill
Dwight D. Kimball
Henry C. Meyer, Jr.
Arthur K. Ohmes
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1917

President....J. Irvine Lyle
1st Vice-President....Arthur K. Ohmes
2nd Vice-President....Fred R. Still
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, J. Irvine Lyle

A. K. Ohmes, *Vice-Chm.*
Homer Addams
Davis S. Boyden
Bert C. Davis
Milton W. Franklin
Charles A. Fuller
Harry M. Hart
E. Vernon Hill
James M. Stannard
Fred R. Still
Walter S. Timmis
Casin W. Obert, *Secy.*

1918

President....Fred R. Still
1st Vice-President....Walter S. Timmis
2nd Vice-President....E. Vernon Hill
Treasurer....Homer Addams
Secretary....Casin W. Obert

Council

Chairman, Fred R. Still

W. S. Timmis, *Vice-Chm.*
Homer Addams
William H. Driscoll
Howard H. Fielding
H. P. Gant
C. W. Kimball
J. Irvine Lyle
E. Vernon Hill
Frank G. Phegley
Fred. W. Powers
Champlain L. Riley
Casin W. Obert, *Secy.*

1919

President............Walter S. Timmis
1st Vice-President............E. Vernon Hill
2nd Vice-President............Milton W. Franklin
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Walter S. Timmis

E. Vernon Hill, *Vice-Chm.*	Frank G. Phegley
Homer Addams	Fred. W. Powers
Howard H. Fielding	Robt. W. Pryor, Jr.
Milton W. Franklin	Champlain L. Riley
Harry E. Gerrish	Fred R. Still
George B. Nichols	Casin W. Obert, *Secy.*

1920

President............E. Vernon Hill
1st Vice-President............Champlain L. Riley
2nd Vice-President............Jay R. McColl
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, E. Vernon Hill

C. L. Riley, *Vice-Chm.*	Jay R. McColl
Homer Addams	George B. Nichols
Jos. A. Cutler	Robt. W. Pryor, Jr.
Wm. H. Driscoll	W. S. Timmis
A. C. Edgar	Perry West
Alfred Kellogg	Casin W. Obert, *Secy.*

1921

President............Champlain L. Riley
1st Vice-President............Jay R. McColl
2nd Vice-President............H. P. Gant
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Champlain L. Riley

Jay R. McColl, *Vice-Chm.*	E. S. Hallett
Homer Addams	E. Vernon Hill
Jos. A. Cutler	Alfred Kellogg
Samuel E. Dibble	E. E. McNair
Wm. H. Driscoll	Perry West
H. P. Gant	Casin W. Obert, *Secy.*

1922

President............Jay R. McColl
1st Vice-President............H. P. Gant
2nd Vice-President............Samuel E. Dibble
Treasurer............Homer Addams
Secretary............Casin W. Obert

Council

Chairman, Jay R. McColl

H. P. Gant, *Vice-Chm.*	L. A. Harding
Homer Addams	E. E. McNair
Jos. A. Cutler	H. J. Meyer
Samuel E. Dibble	C. L. Riley
Wm. H. Driscoll	Perry West
E. S. Hallett	Casin W. Obert, *Secy.*

1923

President............H. P. Gant
1st Vice-President............Homer Addams
2nd Vice-President............E. E. McNair
Treasurer............Wm. H. Driscoll
Secretary............C. W. Obert

Council

Chairman, H. P. Gant

Homer Addams, *Vice-Chm.*	E. S. Hallett
W. H. Carrier	Alfred Kellogg
J. A. Cutler	Thornton Lewis
S. E. Dibble	E. E. McNair
Wm. H. Driscoll	Perry West

Casin W. Obert, *Secy.*

1924

President............Homer Addams
1st Vice-President............S. E. Dibble
2nd Vice-President............William H. Driscoll
Treasurer............Perry West
Secretary............F. C. Houghten

Council

Chairman, Homer Addams

S. E. Dibble, *Vice-Chm.*	W. E. Gillham
F. Paul Anderson	L. A. Harding
W. H. Carrier	Alfred Kellogg
J. A. Cutler	Thornton Lewis
William H. Driscoll	Perry West
H. P. Gant	F. C. Houghten, *Secy.*

1925

President............S. E. Dibble
1st Vice-President............Wm. H. Driscoll
2nd Vice-President............F. Paul Anderson
Treasurer............Perry West
Secretary............F. C. Houghten

Council

Chairman, S. E. Dibble

Wm. H. Driscoll, *Vice-Chm.*	W. T. Jones
Homer Addams	Thornton Lewis
F. Paul Anderson	J. H. Walker
W. H. Carrier	Perry West
J. A. Cutler	A. C. Willard
W. E. Gillham	F. C. Houghten, *Secy.*

1926

President............W. H. Driscoll
1st Vice-President............F. Paul Anderson
2nd Vice-President............A. C. Willard
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, W. H. Driscoll

F. Paul Anderson, *Vice-Chm.*	C. V. Haynes
W. H. Carrier	W. T. Jones
J. A. Cutler	E. B. Langenberg
S. E. Dibble	Thornton Lewis
W. E. Gillham	J. F. McIntire

A. C. Willard

1927

President............F. Paul Anderson
1st Vice-President............A. C. Willard
2nd Vice-President............Thornton Lewis
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, F. Paul Anderson

A. C. Willard, *Vice-Chm.*
H. H. Angus
W. H. Carrier
W. H. Driscoll
Roswell Farnham
H. H. Fielding
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
Thornton Lewis
J. F. McIntire
H. Lee Moore
F. B. Rowley

1928

President............A. C. Willard
1st Vice-President............Thornton Lewis
2nd Vice-President............L. A. Harding
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson

Council

Chairman, A. C. Willard

Thornton Lewis, *Vice-Chm.*
F. Paul Anderson
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
J. J. Kissick
E. B. Langenberg
J. F. McIntire
H. Lee Moore
F. B. Rowley

1929

President............Thornton Lewis
1st Vice-President............L. A. Harding
2nd Vice-President............W. H. Carrier
Treasurer............W. E. Gillham
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, Thornton Lewis

L. A. Harding, *Vice-Chm.*
H. H. Angus
W. H. Carrier
N. W. Downes
Roswell Farnham
W. E. Gillham
C. V. Haynes
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
W. A. Rowe
F. B. Rowley
A. C. Willard

1930

President............L. A. Harding
1st Vice-President............W. H. Carrier
2nd Vice-President............F. B. Rowley
Treasurer............C. W. Farrar
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, L. A. Harding

W. H. Carrier, *Vice-Chm.*
H. H. Angus
D. S. Boyden
R. H. Carpenter
J. D. Cassell
N. W. Downes
Roswell Farnham
C. W. Farrar
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
Thornton Lewis
F. C. McIntosh
W. A. Rowe
F. B. Rowley

1931

President............W. H. Carrier
1st Vice-President............F. B. Rowley
2nd Vice-President............W. T. Jones
Treasurer............F. D. Mensing
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, W. H. Carrier

F. B. Rowley, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
J. D. Cassell
E. O. Eastwood
Roswell Farnham
E. H. Gurney
L. A. Harding
John Howatt
W. T. Jones
E. B. Langenberg
G. L. Larson
F. C. McIntosh
F. D. Mensing
W. A. Rowe

1932

President............F. B. Rowley
1st Vice-President............W. T. Jones
2nd Vice-President............C. V. Haynes
Treasurer............F. D. Mensing
Secretary............A. V. Hutchinson
Technical Secretary............P. D. Close

Council

Chairman, F. B. Rowley

W. T. Jones, *Vice-Chm.*
D. S. Boyden
E. K. Campbell
R. H. Carpenter
W. H. Carrier
John D. Cassell
E. O. Eastwood
Roswell Farnham
F. E. Giesecke
E. Holt Gurney
C. V. Haynes
John Howatt
G. L. Larson
J. F. McIntire
F. D. Mensing
W. E. Stark

1933

President............W. T. Jones
1st Vice-President............C. V. Haynes
2nd Vice-President............John Howatt
Treasurer............D. S. Boyden
Secretary............A. V. Hutchinson

Council

Chairman, W. T. Jones

C. V. Haynes, *Vice-Chm.*
R. H. Carpenter
F. D. Cassell
D. S. Boyden
John Howatt
F. C. McIntosh
L. W. Moon
F. E. Giesecke
G. L. Larson
J. F. McIntire
W. E. Stark
E. K. Campbell
E. O. Eastwood
R. Farnham
E. H. Gurney
F. B. Rowley